INTERACTIVE COMPUTER AIDED DESIGN OF STEERING SYSTEMS

by

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MAHESH GUPTA

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CERTIFICATE

It is to certify that the thesis entitled "Interactive Computer Aided Design of Steering Systems" is a record of work carried out by Mahesh Gupta under our supervision and that it has not been submitted elsewhere for a degree.

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NOMENCLATURE

$\underline{a}, \dot{\underline{a}}, \ddot{\underline{a}}$	: Position, velocity and acceleration vectors of the spherical joint between the coupler and the input link of RSSR linkage.
$\underline{a}_0$	: Position vector of the revolute joint between the input and the fixed links of RSSR linkage.
$\underline{a}_1$	: Position vector of the spherical joint between the coupler and the input link of RSSR linkage in the initial configuration.
b	: Wheel base.
$\underline{b}, \dot{\underline{b}}, \ddot{\underline{b}}$	: Position velocity and acceleration vectors of the spherical joint between the coupler and the output link of RSSR linkage.
$\underline{b}_0$	: Position vector of the revolute joint between the output and the fixed links of RSSR linkage.
$\underline{b}_1$	: Position vector of the spherical joint between the coupler and the output link of RSSR linkage in the initial configuration.
b_t	: Tire width.
d	: Distance between the two king pins
d_s	: Diameter of steering wheel

e	:	King pim offset
f_i	:	Internal force applied by i^{th} link of RSSR linkage on $(i-1)^{th}$ link.
h	:	Nut displacement
h_t	:	Effective torque arm for turning a tire
k_i	:	$\dfrac{\text{Distance from } (i-1)^{th} \text{ link to the point of application of external force on } i^{th} \text{ link}}{\text{Length of } i^{th} \text{ link of RSS linkage}}$
k_t	:	Polar radius of gyration of the contact region between road and a tire.
l	:	Perpendicular distance from the center axis of a vehicle to the point of intersection of the axes of the two front wheels.
l_s	:	Lead of the screw
$\underline{m}_1$	:	Moment applied by the input link of RSSR linkage on the fixed link.
m_{1a}	:	Component of $\underline{m}_1$ along the axis of rotation of the input link of RSSR linkage.
$\underline{m}_4$	:	Moment applied by the fixed link on the output link of RSSR linkage.
m_{4a}	:	Component of $\underline{m}_4$ along the axis of rotation of the output link of RSSR linkage.
$\underline{r}^{ba}$	:	$\underline{b} - \underline{a}$
$\underline{r}^{b_o b}$	:	$\underline{b}_o - \underline{b}$
$\underline{u}_a$	:	Unit vector along the axis of rotation of the input link of RSSR linkage.

$\underset{\sim}{u}_b$: Unit vector along the axis of rotation of the output link of RSSR linkage.

$\underset{\sim}{u}_{lk}$: Unit vector along the left king pin.

$\underset{\sim}{u}_{ls}$: Unit vector along the axis of rotation of the left tire in a steered configuration.

$\underset{\sim}{u}_{lt}$: Unit vector along the axis of rotation of the left tire, when the vehicle is moving on a straight path.

$\underset{\sim}{u}_{rk}$: Unit vector along the right king pin.

$\underset{\sim}{u}_{rs}$: Unit vector along the axis of rotation of the right tire in a steered configuration.

$\underset{\sim}{u}_{rt}$: Unit vector along the axis of rotation of the right tire, when the vehicle is moving on a straight path.

w_t : Load on a steered wheel.

$\underset{\sim}{A}_d$: Acceleration of the rack.

A_{err} : Area steering error.

D : Length of the spindle arm.

$\underset{\sim}{D}_d$: Displacement of the rack.

D_p : Pitch diameter of the pinion.

F_i : External force on the i^{th} link of RSSR linkage.

F_s : Steering effort

M_{apm} : Axial moment required to turn the Pitman arm.

M_k : Torque required to turn a stationary wheel.

R : Constant gear ratio.

S_i : Initial nut displacement.

$\underline{V}_d$	:	Velocity of the rack.
X_{err}	:	Distance steering error.
$\alpha, \dot{\alpha}, \ddot{\alpha}$	:	Angular displacement, velocity and acceleration of the input link of RSSR linkage.
α_1, α_2	:	Angular acceleration of the driving and driven yokes of a universal joint.
α_d	:	Angular acceleration of the Pitman arm.
α_{gin}	:	Angular acceleration of the input shaft of steering gear box.
α_s	:	Angular acceleration of screw.
$\beta, \dot{\beta}, \ddot{\beta}$	:	Angular displacement, velocity and acceleration of the output link of RSSR linkage.
γ	:	Angle between the driving and driven yokes. of universal joint.
μ	:	Effective friction coefficient between road and a tire.
η_g	:	Efficiency of gear box.
θ_1, θ_2	:	Angular rotation of the driving and the driven yokes of universal joints.
θ_c	:	Camber angle.
θ_d	:	Angular rotation of the Pitman arm
θ_{di}	:	Total rotation of the spindle arm
θ_{err}	:	Angle steering error.
θ_{gin}	:	Angular rotation of the input shaft of steering gear box.

θ_i	:	Initial rotation of the spindle arm.
θ_l	:	Angular rotation of the left tire.
θ_{pl}	:	Angle between the Z axis and the projection of $\underline{U}_{ls}$ in XZ plane.
θ_{pr}	:	Angle between the Z axis and the projection of $\underline{U}_{rs}$ in XZ plane.
θ_r	:	Angular rotation of the right tire.
θ_s	:	Angular rotation of screw.
ϕ_1 , ϕ_2	:	Initial angular rotation of the driving and the driven yokes of universal joints.
ω_1 , ω_2	:	Angular velocities of the driving and the driven yokes of universal joints.
ω_d	:	Angular velocity of the Pitman arm.
ω_{gin}	:	Angular velocity of the input shaft of steering gear box.
ω_s	:	Angular velocity of screw.
ω_{st}	:	Angular velocity of steering.

ABSTRACT

In the present work, various types of steering systems, used in commercial vehicles, have been analysed using the technique of interactive computer aided design. Vector and matrix methods have been used for kinematic and force analyses. The program also calculates the steering error and steering effort - hand force required to steer the vehicle.

Steering system of a vehicle consists of three types of mechanisms - universal joints between steering wheel and steering gear box, steering gear box and steering linkage to transmit the motion from the gear box to the two steered tires. Many options are available for each of these constituent mechanisms. But basic building mechanisms being same, all types of steering systems have been analyzed using a few subroutines, which are called repeatedly.

Variations in various parameters, with steering wheel rotation, have been studied. The study helps in evaluating the performance of a steering system. The desirable trends of these parameters have also been presente

Chapter 1

INTRODUCTION

1.1 Role of Steering System in a Vehicle

A vehicle is a machine for carrying and transporting things. To follow a desired route and also to avoid the obstacle, a vehicle is expected to change its direction every now and then. Basic purpose of a steering system is to make it possible for the driver to direct the course of the vehicle. Though, steering system is generally understood to be only for directing the course of the vehicle, another very important role a steering system plays is to maintain the directional stability regardless of the irregularities in the surface over which the vehicle is moving.

1.2 Basics of Vehicle Steering

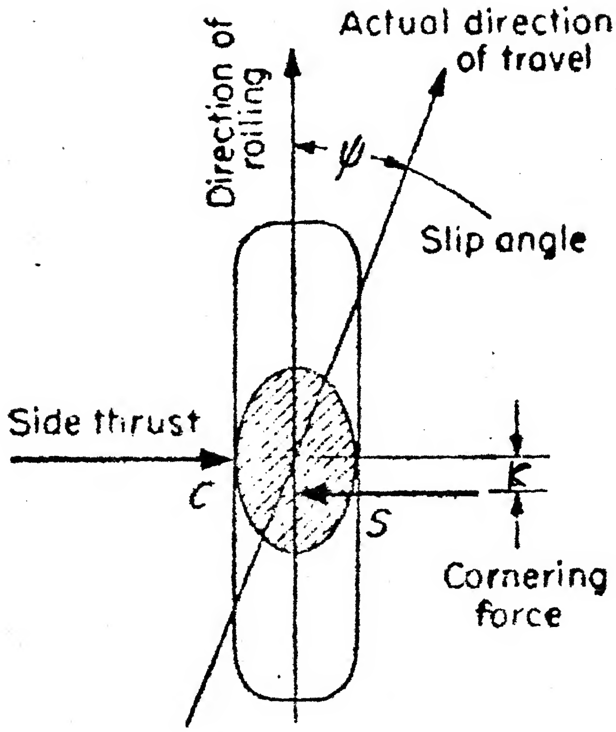
Just like any other body moving on a circular path, a vehicle on a turn requires a centripetal force directed towards the center of the arc on which the vehicle is moving. As the only external force on the vehicle is between its tires and the road (neglecting the air resistance), friction between the tires

and the road should provide enough force which can balance the centrifugal force, to avoid the vehicle from flying off its course.

Whenever the actual direction of travel of a vehicle differs from its longitudinal direction (the angle between the two is called slip angle), a side thrust is developed along the axis of the tire. The rubber tire counter-acts this with frictional reaction called cornering force, in the direction of its axis of rotation (Fig. 1.1). It is the cornering force, which provides the necessary centripetal force for vehicle going on a turn. Magnitude of the cornering force increases with slip angle. Therefore, with the increasing sharpness of the turn, wheels should make larger and larger angle with the longitudinal plane of the vehicle to generate the sufficient cornering force.

1.3 Steering Error

When a vehicle is moving on a straight path, all the wheels roll about their axes of rotation. On a turn, the motion of a vehicle is not likely to be of pure rolling, but that of sliding and rolling. Any sliding of the vehicle is detrimental, particularly from the view point of tyre wear. On a turn, the condition of pure rolling is achieved only if the axes of



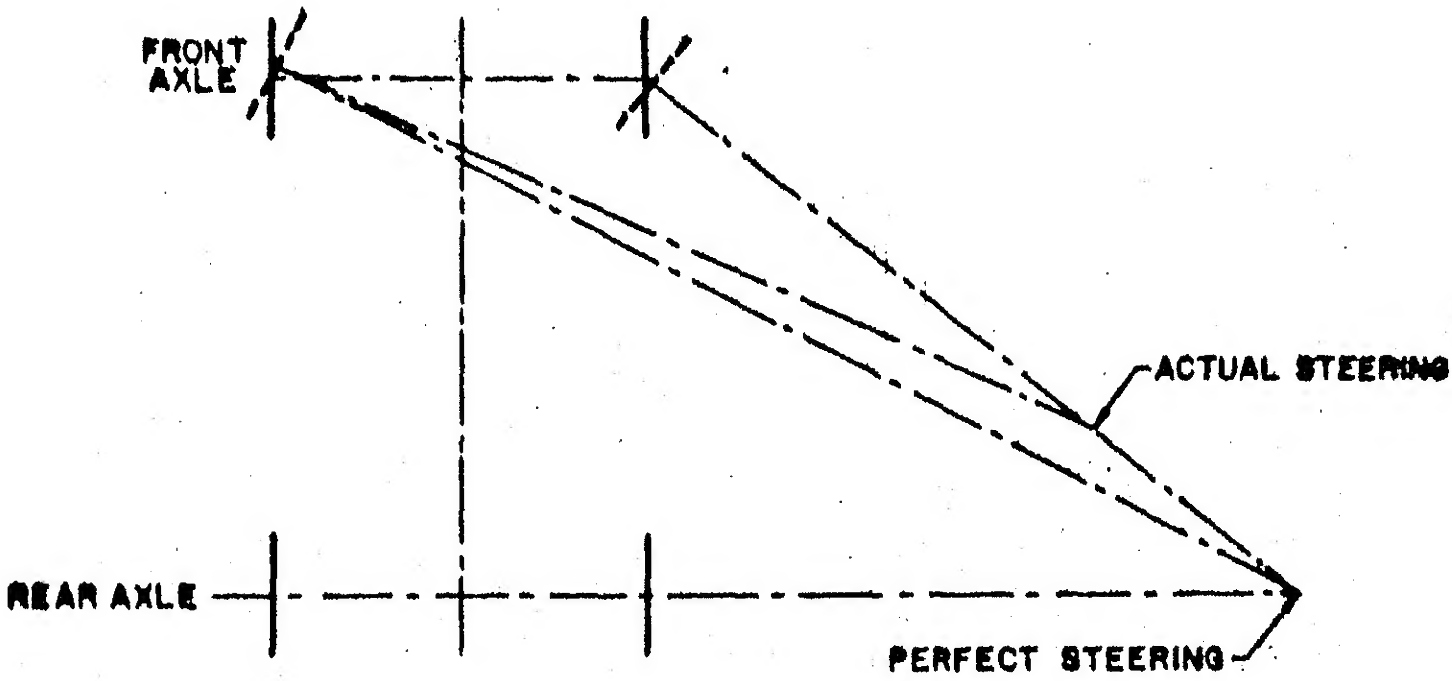
Fig·1·1 Side thrust and the cornering force in a rubber tire·

all the wheels intersect at a common point. As the axis of rear wheels is fixed, two front wheels should turn through different angles as to intersect at the common axis of rotation of rear wheels. If the perfect conditions are not met the vehicle will start sliding. This error in steering is termed as steering error (Fig. 1.2).

1.4 Types of Steering Systems

Various types of steering systems are used to achieve the turning of the two front wheels through different angles. Following features are desired from a steering system:

1. Small steering error.
2. Steering effort should be as low as possible. Many times when the effort required is large, power steering is used. In power steering a booster arrangement is provided, which is set into operation when the steering wheel is turned. The booster then takes over and does most of the work.
3. Cramp angle - the angle through which a wheel can be turned - should be sufficiently large, without exceeding practical limitation on linkage size. Steering system of a vehicle (Fig. 1.3) consists of Hooke joint(s), steering gear box and steering linkage.



Fig·1·2 Actual and perfect steering

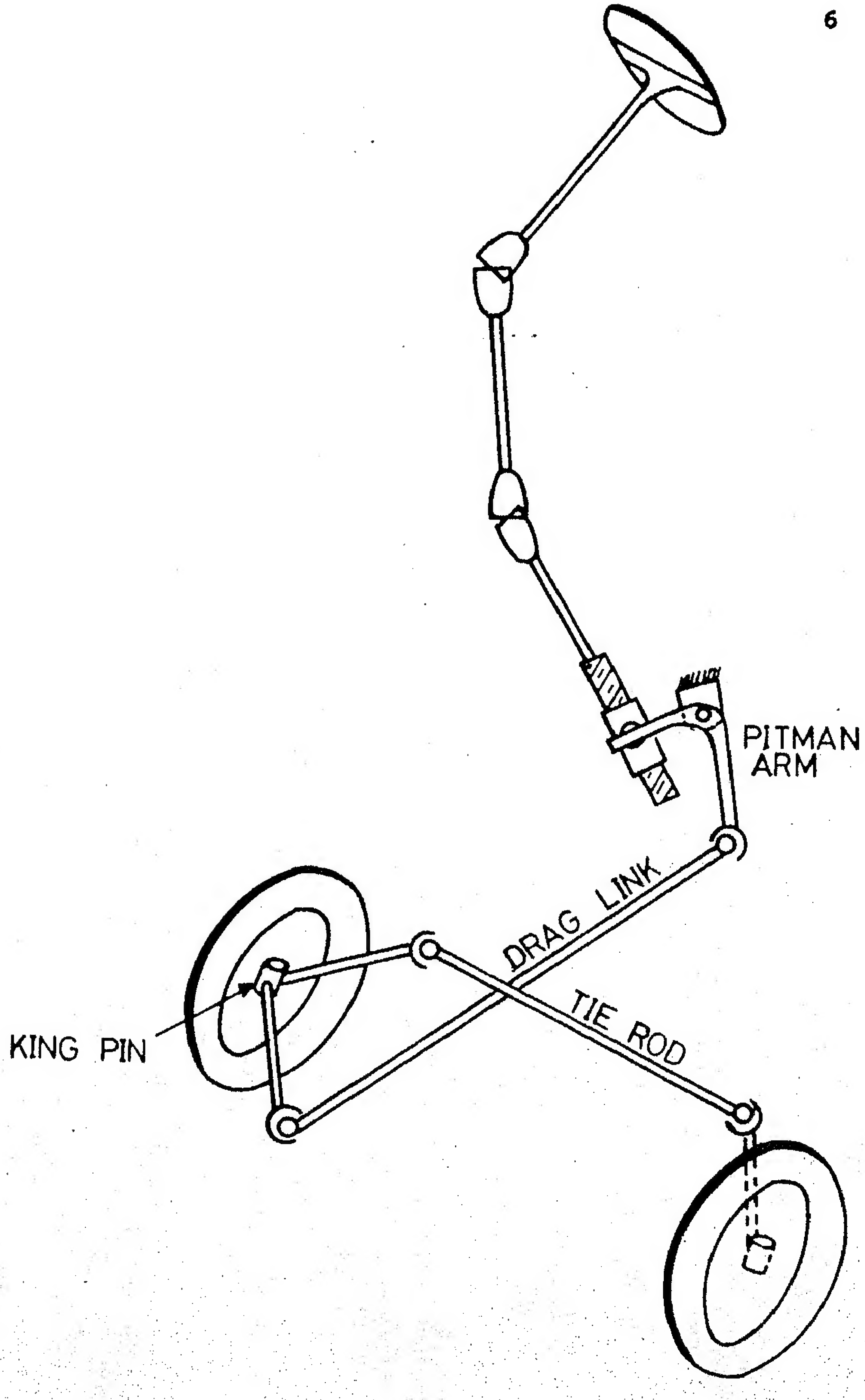


Fig.1.3 Steering system of a truck.

1.4.1 Hooke Joint

Whenever the direction of steering wheel axis differs from the input axis of steering gear box, one or two universal joints are required to connect the two. Two universal joints give the additional advantages of constant velocity ratio being possible and also provide an easy way for making steering shaft collapsible.

1.4.2 Steering Gear Box

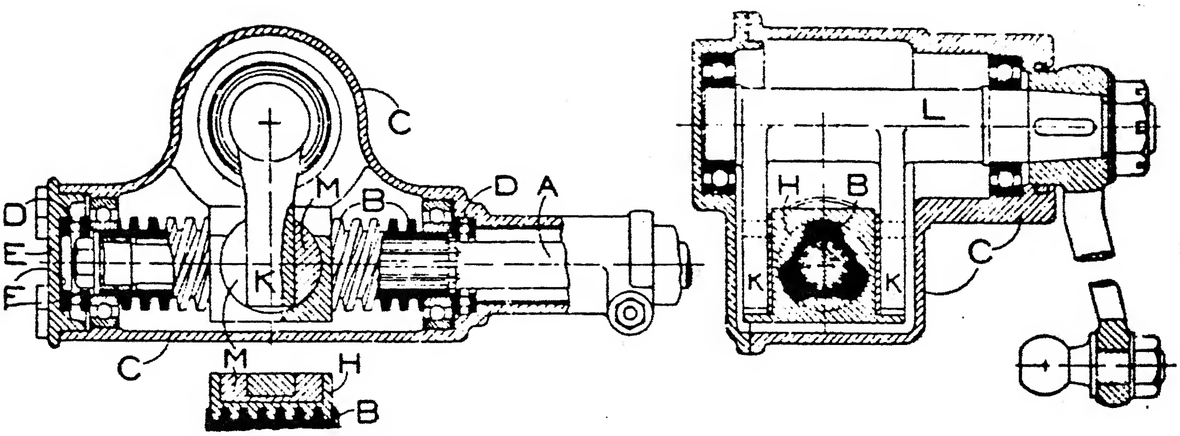
For steering gear box mainly three types of mechanisms are used:

(i) Screw and nut steering mechanism

(ii) Constant ratio steering mechanism

(iii) Cam steering mechanism.

An example of screw and nut type of steering gear is shown in(Fig. 1.4). A multiple threaded screw is attached to the output shaft of the universal joint or directly to the steering shaft if no universal joint is used. This screw rotates in a cast iron casing. A nut moves axially on this as the screw is rotated. This motion is caused to rotate the spindle arm, on which the Pitman arm is attached. As the nut moves in a straight line while the spindle arm moves in a circular path to allow the motion of both, two pads are provided in



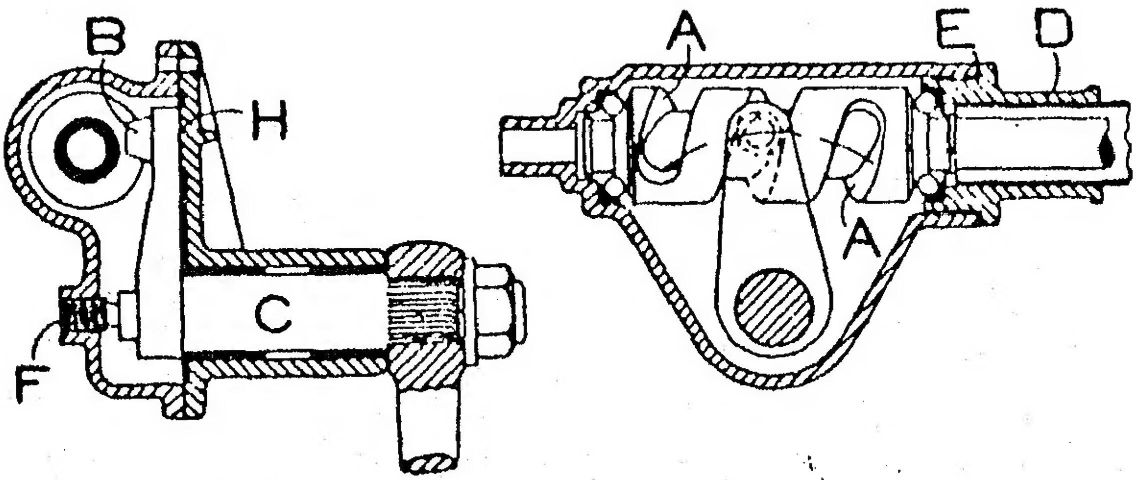
A-Steering shaft.
B-Screw.
C-Cast iron casing.
D-Thrust bearings.
E-Cover plate.
F-Washer to eleminate end play.
H-Nut.
K-Spindle arm.
L-Spindle.
M-Bronze pads.

Fig. 1·4 Screw & nut steering gear mechanism.

cylindrical recesses formed in the side of nut. These pads are provided with parallel grooves to receive the arm of the spindle and are free to rotate in the recesses.

In constant ratio type steering gear a pair of helical bevel gears (or any other similar mechanism) is used to get the angular velocity in the direction of the axis of rotation of the Pitman arm. Magnitude of the angular velocity is also reduced. Another example of constant ratio type steering gear is rack and pinion mechanism. It has a pinion at the end of the steering shaft, which is meshed with a rack housed in a tubular casing. When the wheel is turned, the pinion rotates and rack moves side-wards. This motion is carried through the tie rods to the steering arms at the front wheels.

Cam steering mechanism (Fig. 1.5) works on the same principle as screw and nut steering mechanism. In cam steering mechanism a groove is cut at the end of the steering shaft. A roller or a lever is meshed in this groove which rotates when the steering shaft is rotated, just like the spindle in the screw and nut mechanism.



A-Helical groove·
B-Lever·
C-Spindle·
D-Clamping screw·

E-Washer to eliminate end play
F- Screw to prevent lever from meshing too deeply with the groove A·
H- Cover plate·

Fig·1·5 Screw & lever cam steering gear mechanism

1.4.3 Steering Linkage

Steering linkage which transmits the motion of the Pitman arm to the two steered wheels, consists of a combination of Revolute - Spherical - Spherical - Revolute (RSSR) and Spherical - Spherical - Revolute (SSR) linkages. (Fig. 1.6, 1.7). Various steering linkages used in practice are shown in (Figs. 1.8 to 1.13). Spindle lever linkage (Fig. 1.8) and center point linkage (Fig. 1.10) are most commonly used. These standard steering linkages are modified depending upon other design requirements. For instance, in case of spindle lever linkage if the drag link is very long then linkage is modified to have an intermediate Pitman arm (Fig. 1.14). With spindle lever linkage if the vehicle has two front axles, then one more RSSR linkage is used to transfer the motion to the Ackerman linkage used for second front axle (Fig. 1.15).

1.5 Directional Stability and Front-End Geometry

As mentioned in the beginning second important function of a steering system is to maintain the directional stability. Directional stability is the tendency of vehicle to maintain a direction close to the desired course under the action of a disturbing force and to return back to the equilibrium configuration after the

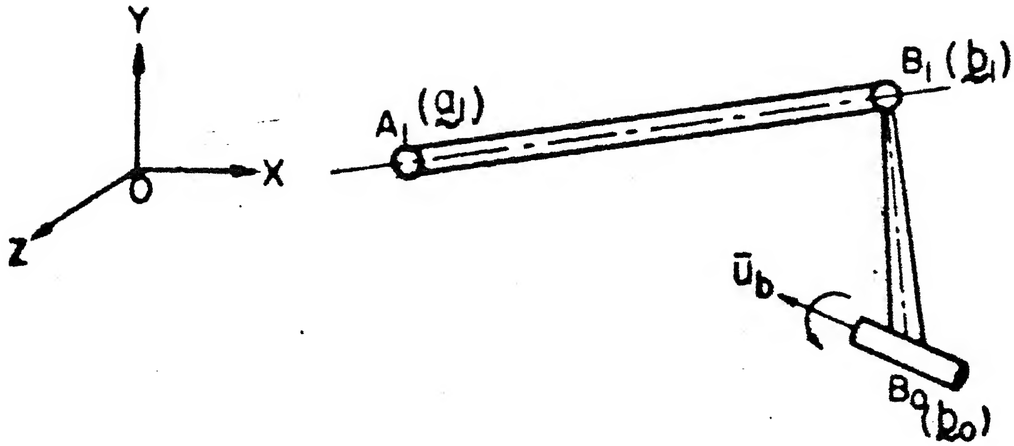


Fig. 1.6 S-S-R Linkage.

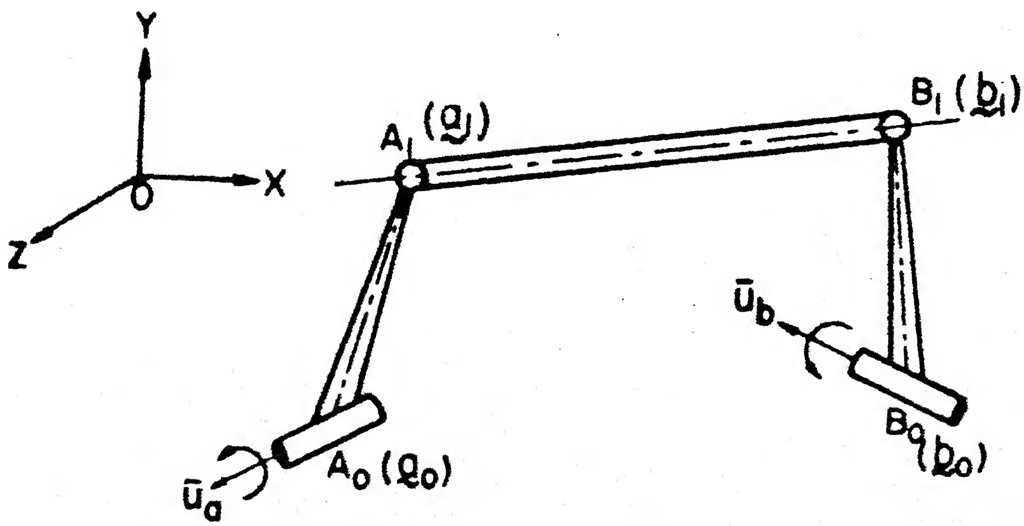
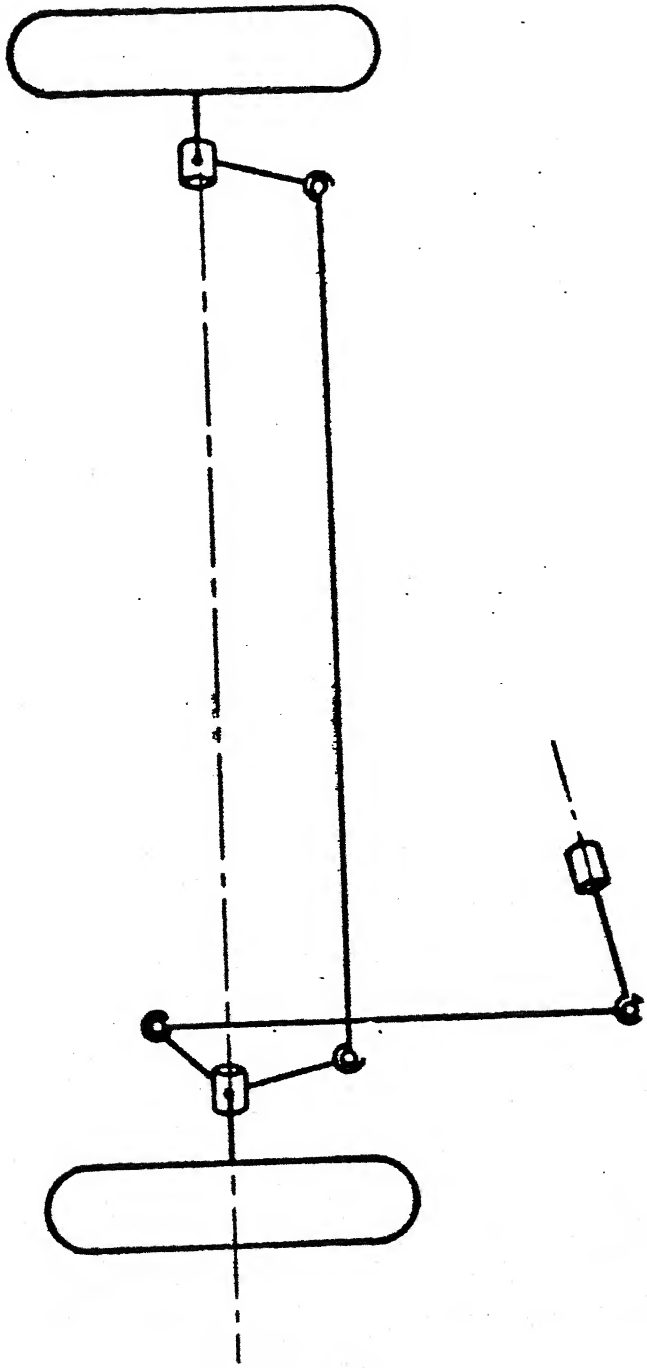
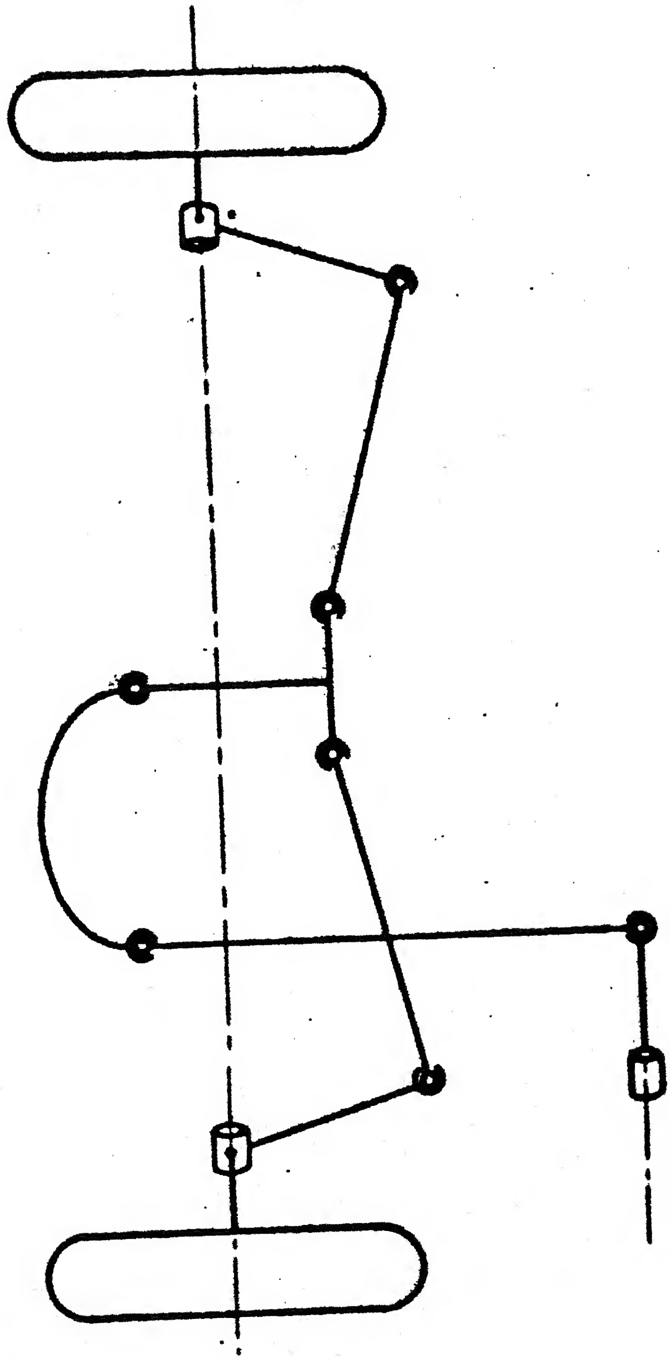


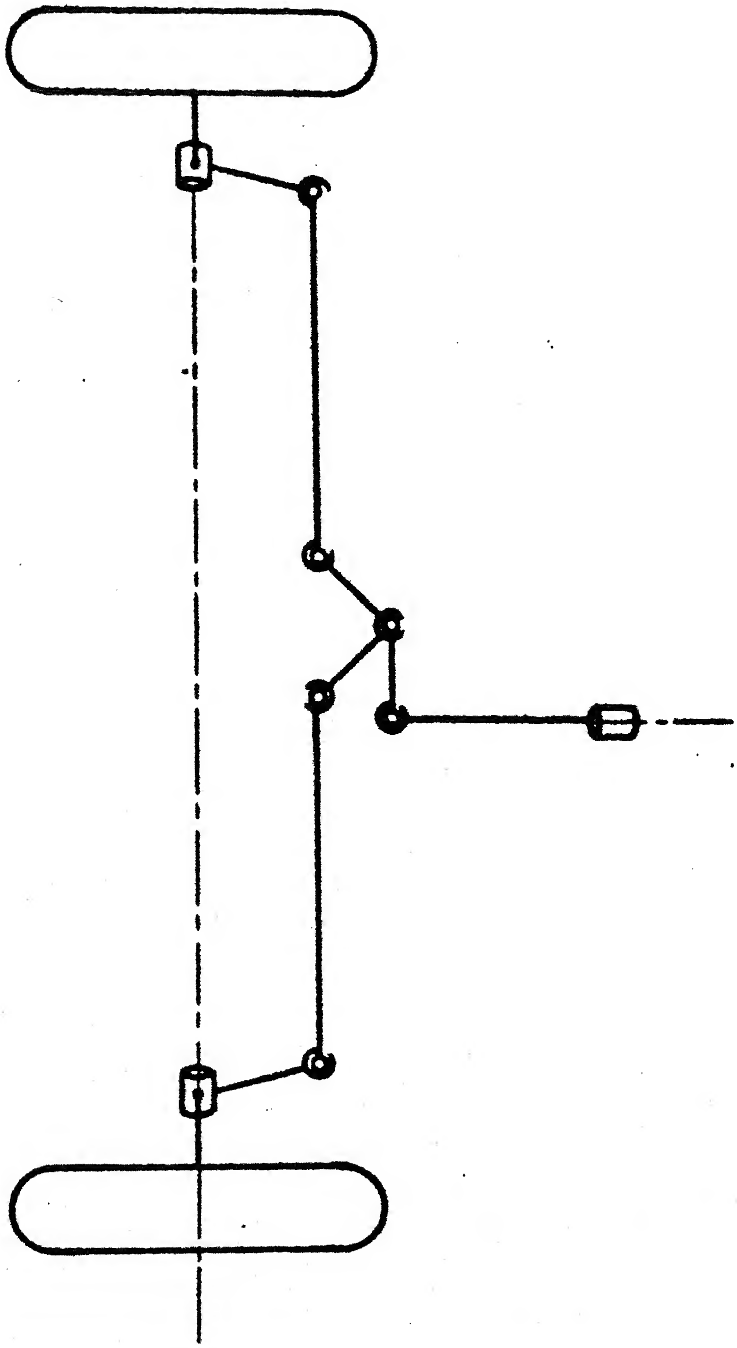
Fig. 1.7 R-S-S-R Linkage.



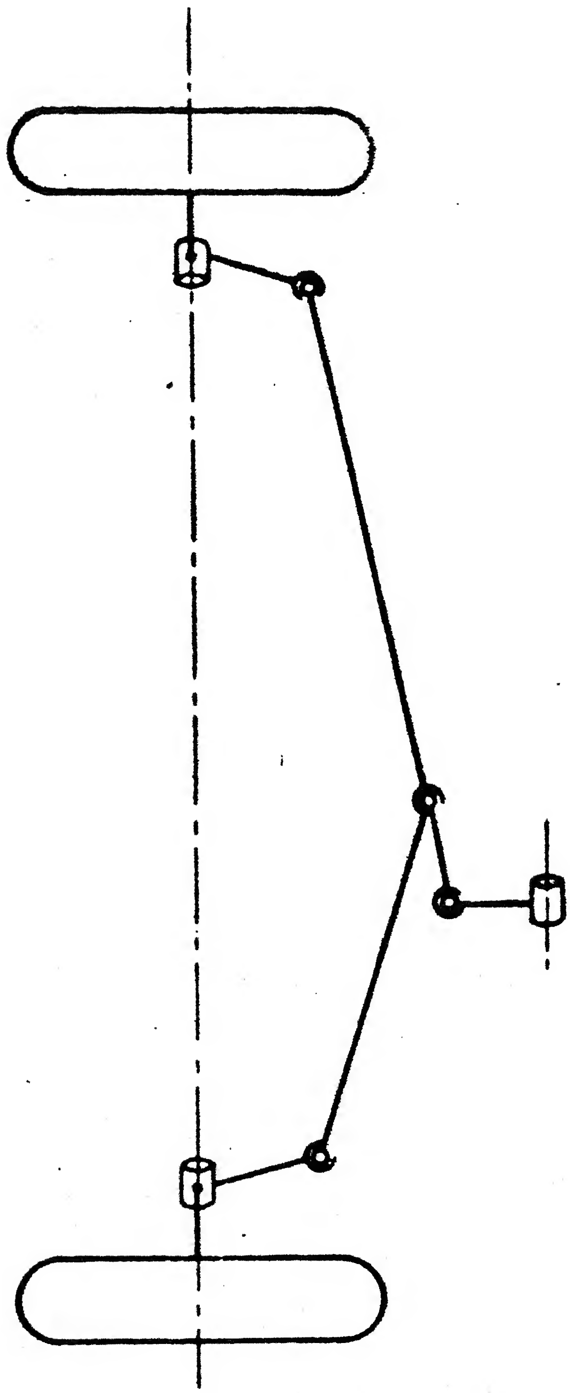
Fig·1·8 Steering - linkage



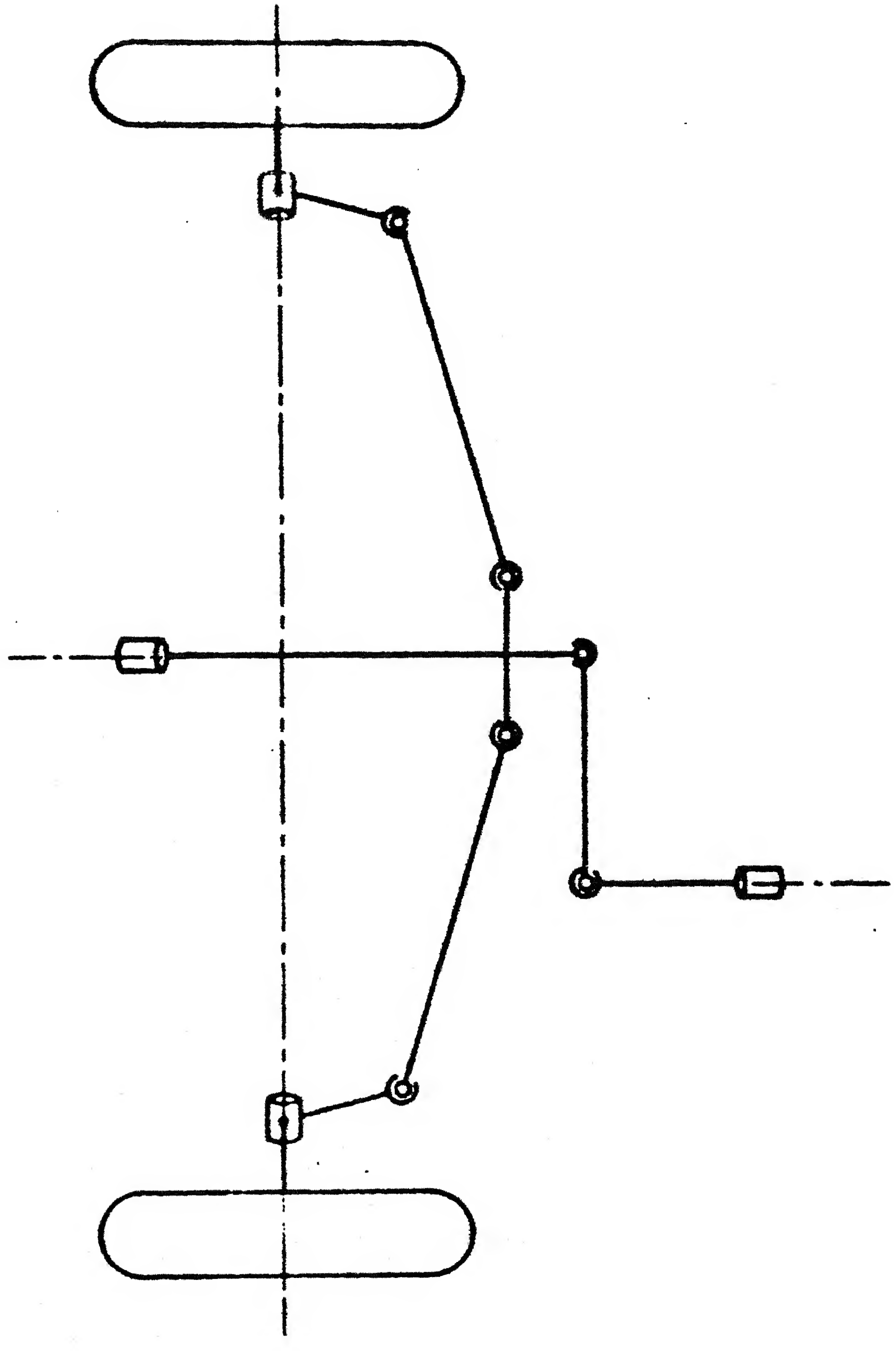
Fig·1·9 Steering-linkage



Fig· 1·10 Steering-linkage



Fig·1·11 Steering - linkage



Fig·1·12 Steering - linkage

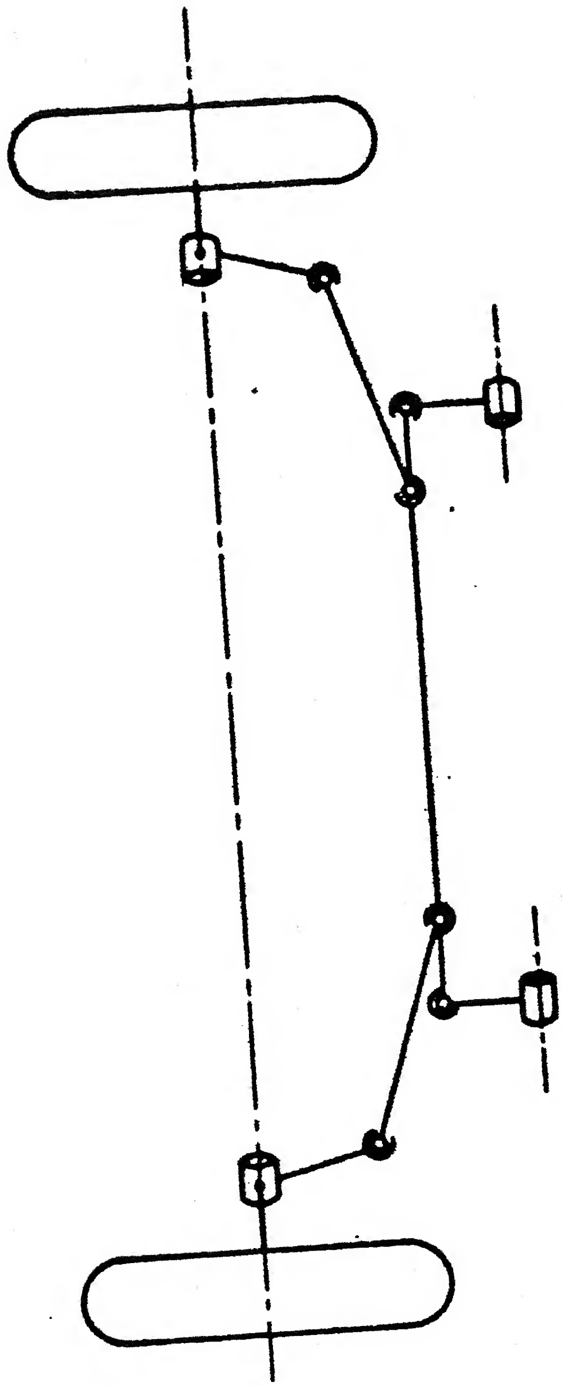
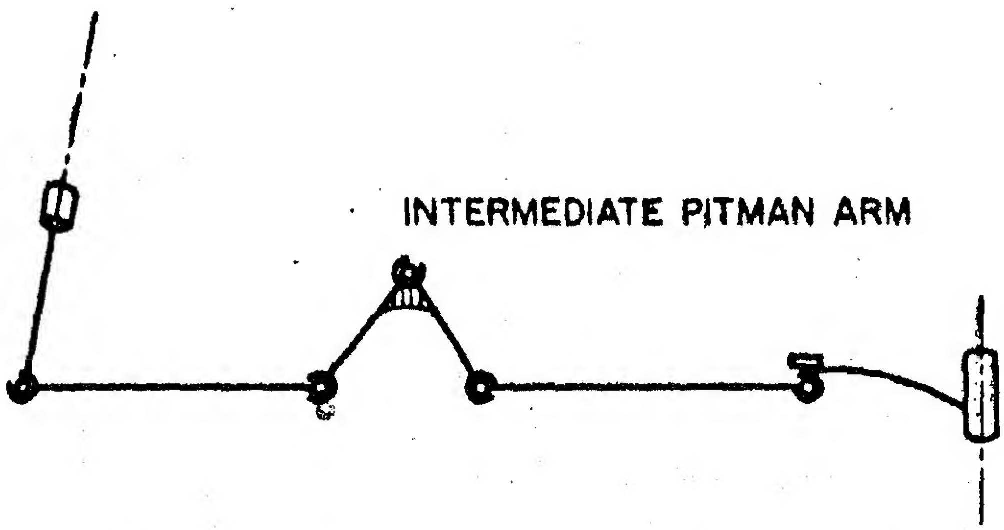


Fig. 1.13 Steering-linkage



Fig·1·14 Steering geometry with intermediate Pitman-arm

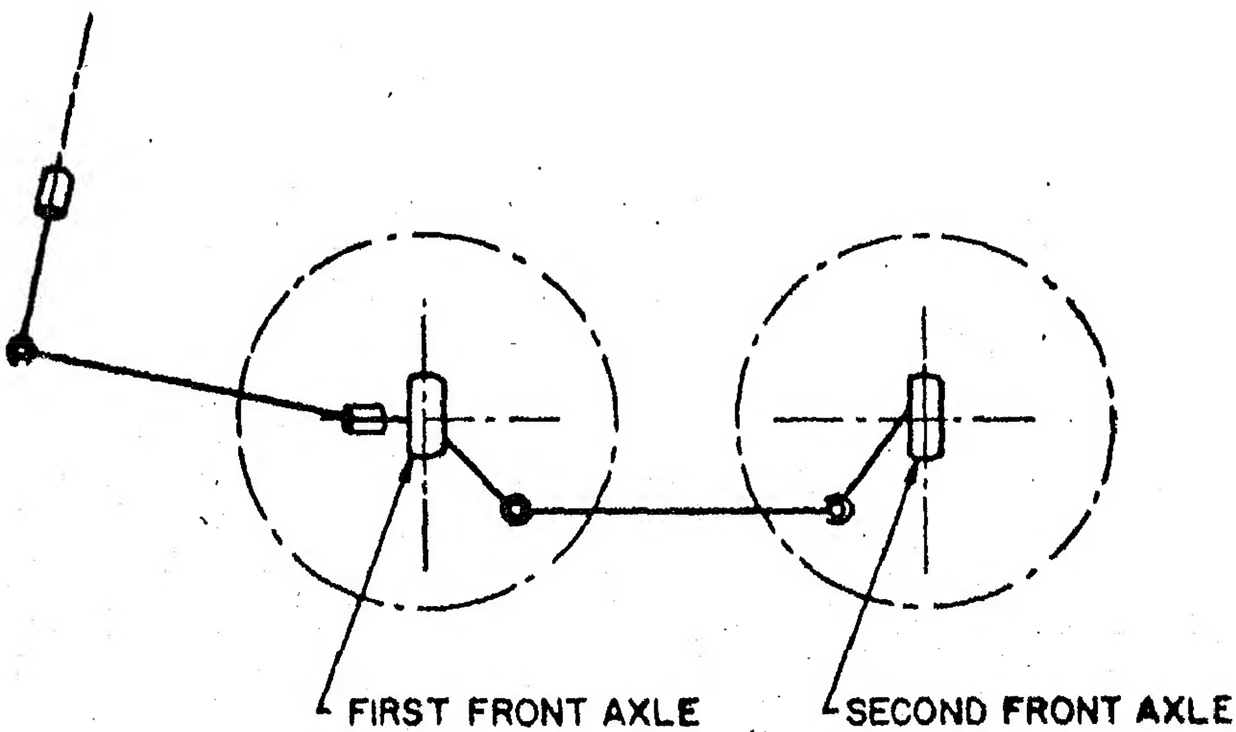


Fig. 1.15 Steering geometry for a vehicle with two front axles

force is removed. For this purpose, small inclinations, discussed below, are given to king pins and to the two front wheels (Fig. 1.16).

1.5.1 King Pin Inclination

King pin inclination is the sideward tilt of king pin from the vertical in transverse plane. The tilt is always inward as shown in Fig. 1.16. Usual king pin inclination is about 6 degrees. It provides directional stability by exerting a self aligning torque to bring the wheel back to the straight ahead position after the turn is over. It also reduces the steering effort by decreasing the king pin offset.

1.5.2 Caster Angle

Caster is the forward or backward tilt of king pin in longitudinal plane. Usual values for caster are between 3 to 5 degrees for front wheel steered vehicle. (For rear wheel steered vehicles, a negative caster angle is used). Main purpose for using positive caster is the self-aligning effect it produces.

1.5.3 Camber Angle

Camber is the sideward tilt of the front wheels from vertical. Usual values of camber are 1/2 to 1 degrees. Besides decreasing the king pin offset distance

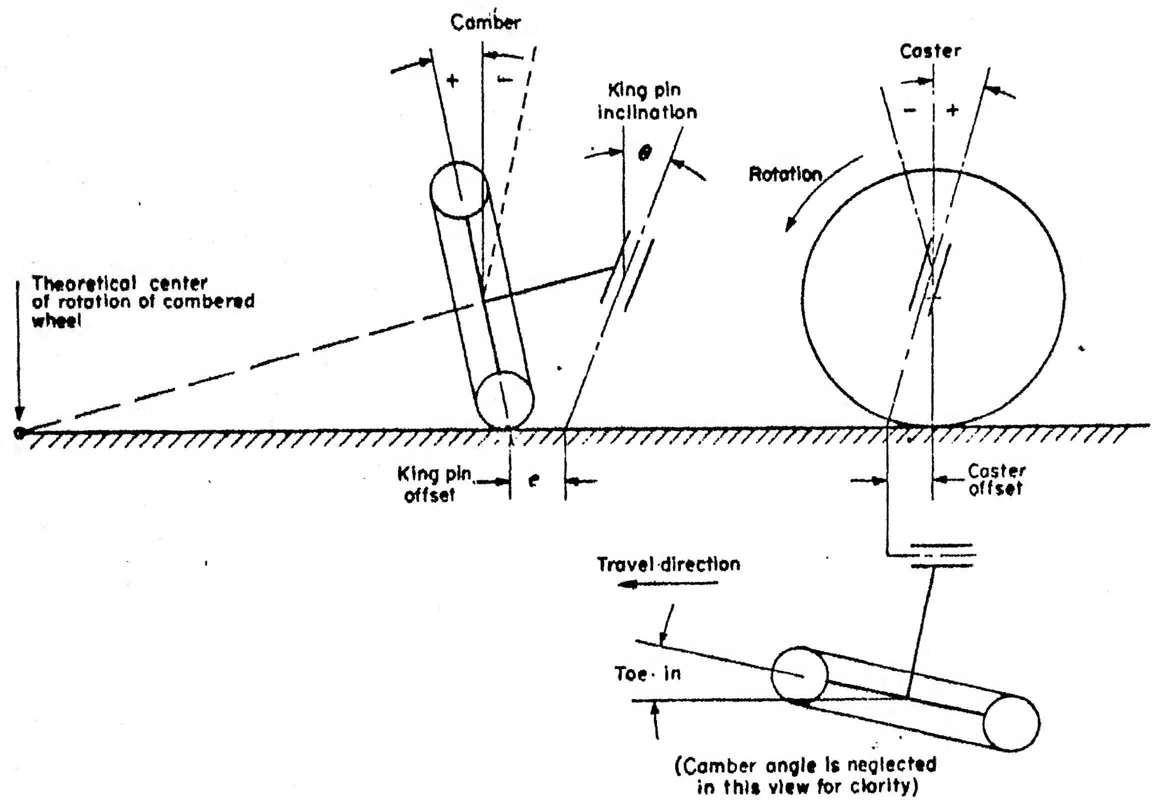


Fig. 1·16 Orientation of wheel and king pin.

(and hence the steering effort), small positive camber prevents a negative camber when the vehicle is heavily loaded. Large value of camber results in excessive tire wear from one side.

1.5.4 Toe-In

Toe-in is the inward tilt of front end of the front wheels. It is measured as the linear deviation of wheel rim, with usual value of about 1/3 cm. Main purpose for toe-in is to create a side thrust capacity for absorbing side shocks, caused by irregularities in the road.

1.6 Scope of Present Work

In the usual design process designer, using his past experience and certain key rules, develops an initial design and then he analyses it to check its performance. If it is not upto the mark, the design is modified and the modified version is analysed again. The process is repeated until a satisfactory design, having the desired characteristics, is achieved. In the present work an attempt has been made to take over from designer, the tedious job of analysing the system after every modification, using the technique of interactive computer aided design. The program performs kinematic and force analyses of the steering system using vector and matrix methods [1]. It also calculates the steering error

and steering effort - hand force required to steer the vehicle. It has provisions for using upto two universal joints, any one of the three types of steering gear-boxes and any one of the various steering linkages being used in commercial vehicles.

Chapter 2

KINEMATIC ANALYSIS OF CONSTITUENT MECHANISMS OF A STEERING SYSTEM

2.1 Hooke Joint

The Hooke joint is shown in Fig. 2.1. It consists of two yokes. One of these is the driving member and another one the driven member. A cross connects the two yokes. During motion points A and B in Fig.2.1 describes a circle in vertical plane perpendicular to the plane of paper and points C and D describe another circle in a plane at an angle γ from the vertical plane. These two path circles on which A and C travel are shown in Fig. 2.2. The configuration, in which the points on driving yoke are at the intersection points of these two path circles, will be called zero - displacement configuration (A_o O C_o). A_1 O C_1 is the initial position of the cross and A_2 O C_2 is the present position.

2.1.1 Notations and Kinematic Equations

γ : Angle between driving and driven shafts

$\emptyset_1$: Initial angular displacement of point A from its position in the zero-displacement configuration

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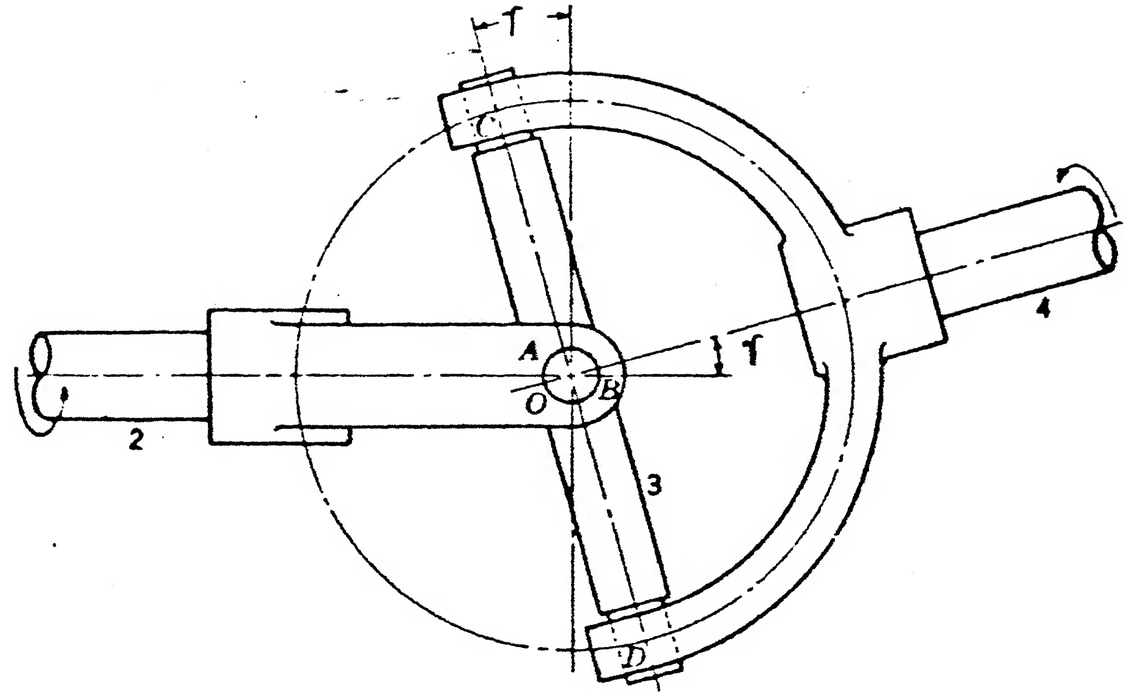


Fig. 2·1 Hooke joint

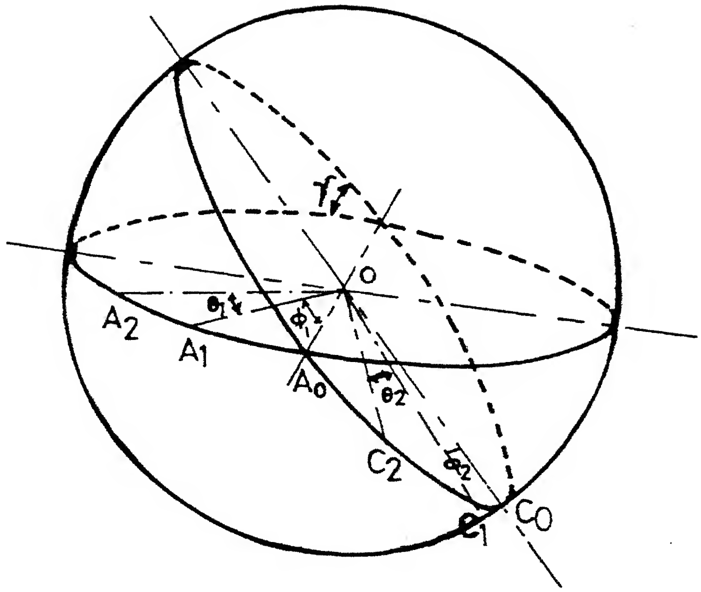


Fig. 2·2 Path circles of point A and point C of the Hooke joint·

ϕ_2 : Initial angular displacement of point C from its position in the zero-displacement configuration

θ_1 : Angular rotation of driving shaft

θ_2 : Angular rotation of driven shaft

ω_1 : Angular velocity of driving shaft

ω_2 : Angular velocity of driven shaft

α_1 : Angular acceleration of driving shaft

α_2 : Angular acceleration of driven shaft

Kinematic equations for Hooke joint are given below:

Using right triangle formula from spherical trigonometry [2]

$$\tan(\theta_2 + \phi_2) = \cos\gamma \,.\, \tan(\theta_1 + \phi_1) \qquad \ldots \qquad (2.1)$$

Differentiating Eqn. 2.1

$$\dot{\theta}_2 \sec^2(\theta_2 + \phi_2) = \dot{\theta}_1 \cos\gamma \sec^2(\theta_1 + \phi_1)$$

$$\omega_2/\omega_1 = \dot{\theta}_2/\dot{\theta}_1 = \frac{\cos\gamma \,.\, \sec^2(\theta_1 + \phi_1)}{\sec^2(\theta_2 + \phi_2)} \qquad \ldots \qquad (2.2)$$

Substituting Eqn. 2.1 in Eqn. 2.2

$$\omega_2/\omega_1 = \frac{\cos\gamma}{1 - \sin^2(\theta_1 + \phi_1) \,.\, \sin^2\gamma} \qquad (2.3)$$

Differentiating Eqn. 2.3

$$\dot{\omega}_2 \, (1 - \sin^2 (\theta_1 + \emptyset_1) \, . \, \sin^2 \gamma) -$$

$$\omega_2 \, (2 \, . \, \sin^2 \gamma \, . \, \sin (\theta_1 + \emptyset_1) . \, \cos (\theta_1 + \emptyset_1)) .$$

$$. \, \omega_1 = \dot{\omega}_1 \, . \, \cos \gamma$$

$$\alpha_2 = \frac{\alpha_1 \cos \gamma + \omega_1 \, . \, \omega_2 \, . \, \sin^2 \gamma \, . \, \sin (2.(\theta_1 + \emptyset_1))}{1 - \sin^2 (\theta_1 + \emptyset_1) \, . \, \sin^2 \gamma} \quad \ldots \quad (2.4)$$

2.1.2 Description of Subroutine Used for Kinematic Analysis of Universal Joint(s) Chain

Name of the subroutine : unanl

Followings are the input variables to the subroutine (Fig. 2.3):

nounjt : Number of universal joints

thetal : Angular rotation of steering wheel

omega 1 : Angular velocity of steering wheel

alpha 2 : Angular acceleration of steering wheel.

If nounjt $\neq$ 0 then following inputs are also required

gama 1 : Angle between the X axis and the projection of the input shaft in XY plane.

gama 2 : Angle between the Z axis and the projection of the input shaft in YZ plane.

gama 3 : Angle between the X axis and the projection of the output shaft in XY plane.

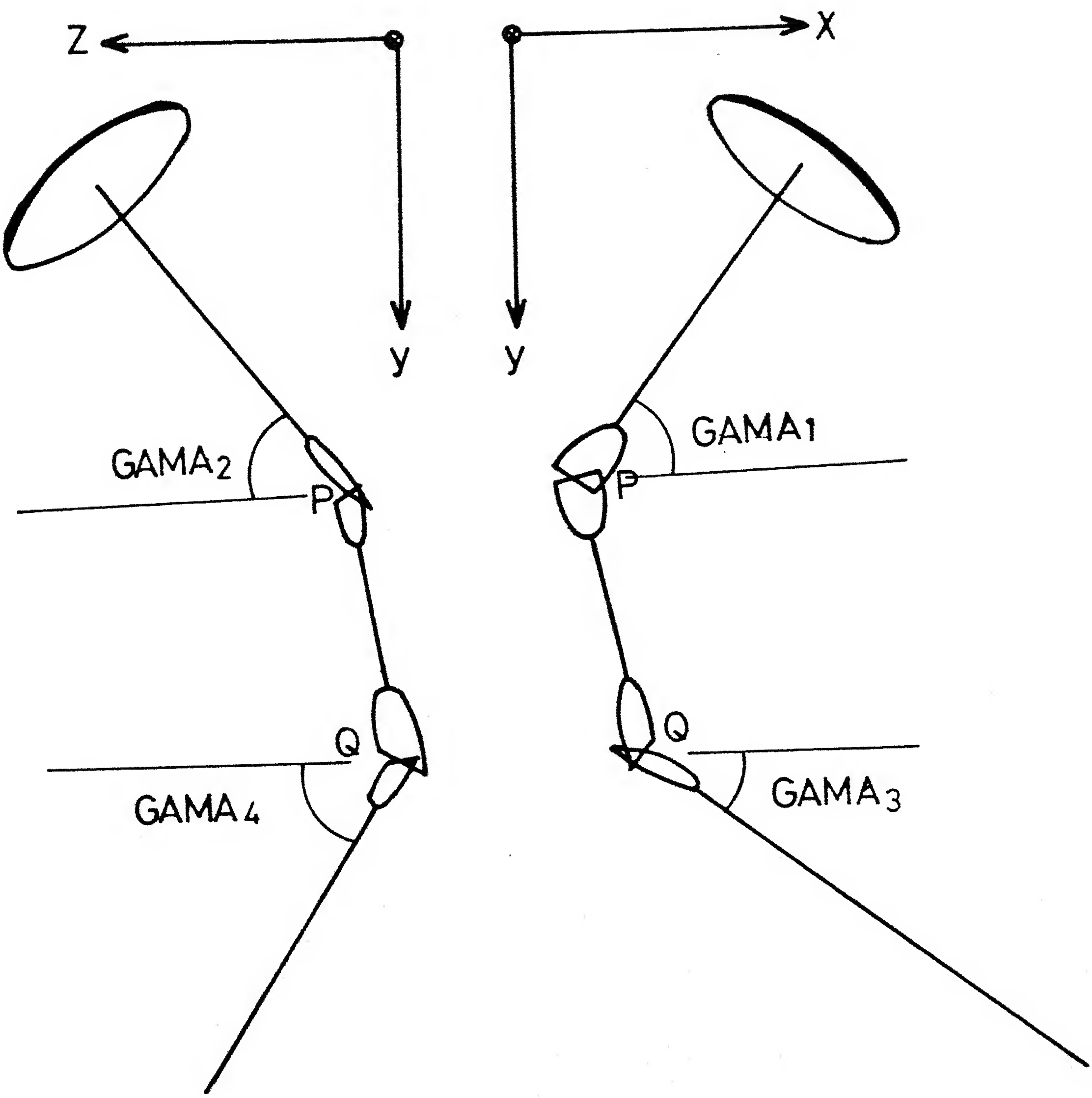


Fig. 2·3 Part of steering system from steering wheel to steering gear box.

gama 1 : Angle between the Z axis and the projection of the output shaft in YZ plane

phy 1 : Initial angular displacement of yoke attached with input shaft from its position in zero-displacement configuration.

If nounjt = 2 then following additional inputs are required

XP,YP,ZP : Coordinates of point P the center point of 'Cross' of the first universal joint

XQ,YQ,ZQ : Coordinates of point Q the center point of 'Cross' of the second universal joint.

Phy 2 : Initial angular displacement of yoke of second universal joint attached to the intermediate shaft from its position in zero-displacement configuration.

Output of the subroutine unanl is given below:

thtout : Angular rotation of output shaft

omgout : Angular velocity of output shaft

alpout : Angular acceleration of output shaft

If number of universal joints is two then following additional output is given

theta2 : Angular rotation of intermediate shaft

omega2 : Angular velocity of intermediate shaft

alpha2 : Angular acceleration of intermediate shaft

Algorithm for unanl is as follows:

```
Unchain ->[ Read (nounjt)
            Case (nounjt) of
            0 : [Read (theta1 ,omega1 ,alpha1 )
                 thtout  =  theta1
                 omgout  =  omega1
                 alpout  =  alpha1 ]
            1 : [Read (gama1 ,gama2 ,gama3 ,gama4 )
                 Calculate the unit vectors along two shaft
                 Calculate the angle between two shafts
                 Read (phy1 )
                 Read (theta1 ,omega1 ,alpha1 )
                 Calculate thtout (Using Eqn. 2.1)
                 Calculate omgout (Using Eqn. 2.3)
                 Calculate alpout (Using Eqn. 2.4)]
```

```
2 : [Read (gama 1 ,gama 2.,gama 3 ,gama 4 )
     Read (XP,YP,ZP,XQ,YQ,ZQ)
     Calculate unit vectors along input,
         intermediate and output shafts
     Calculate angle between input and
         intermediate shafts
     Read (phy1 )
     Read (theta1., omega1 , alpha1 )
     Calculate theta2  (Using Eqn. 2.1)
     Calculate omega2  (Using Eqn. 2.3)
     Calculate alpha2  (Using Eqn. 2.4)
     Read (phy 2 )
     Calculate angle between intermediate
         and output shafts
     Calculate thtout (Using Eqn. 2.1)
     Calculate omgout (Using Eqn. 2.3)
     Calculate alpout (Using Eqn. 2.4)]
                                 → unanl
```

2.2 Steering Gear Box

As described in Section 1.2 three types of stearing gears generally used in practice are:

(i) Screw and nut steering gear mechanism

(ii) Constant ratio steering gear mechanism

(iii) Cam steering gear mechanism.

Kinematic analysis of screw and nut, mechanism and cam steering gear mechanism give same set of equations, whereas rack and pinion and other types of constant ratio mechanisms provide two different sets of equations.

2.2.1 Notations and Kinematic Equations

2.2.1.1 Screw and Nut Steering Mechanism

(Refer Fig. 2.4)

l_s : Lead of the screw

θ_s : Angular rotation of the screw

ω_s : Angular velocity of the screw

α_s : Angular acceleration of the screw

h : Nut displacement

S_i : Initial nut displacement

D : Length of the spindle arm

θ_i : Initial rotation of the spindle arm

θ_{di} : Total rotation of the spindle arm

θ_d : Angular rotation of drop arm

ω_d : Angular velocity of drop arm

α_d : Angular acceleration of drop arm

Kinematic equations for screw and nut type steering gear mechanism are given below:

$$h = l_s \theta_s / 2\pi \tag{2.5}$$

$$\tan \theta_i = S_i / D \tag{2.6}$$

$$\tan \theta_{di} = (h + S_i) / D \tag{2.7}$$

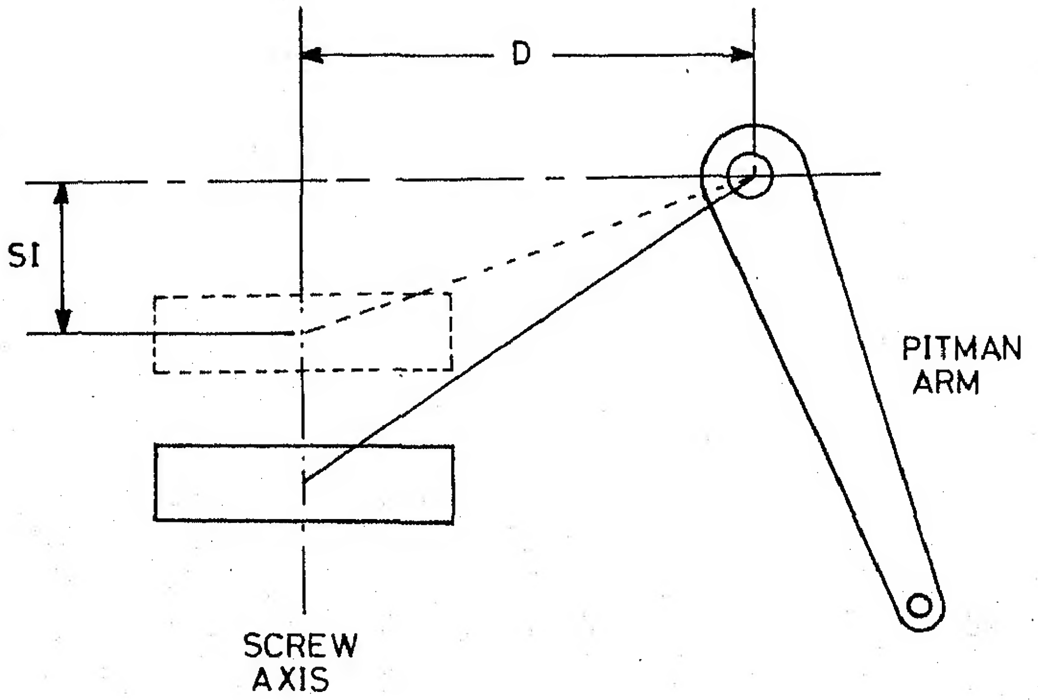


Fig. 2·4 Screw & nut steering gear mechanism.

$$\theta_d = \theta_{di} - \theta_i$$

Using Eqns.(2.6)and(2.7)

$$\theta_d = \tan^{-1}((h + S_i)/D) - \tan^{-1}(S_i/D) \quad (2.8)$$

Differentiating Eqn.(2.7)

$$\omega_d = \dot{\theta}_d = \dot{\theta}_{di} = \frac{\cos^2(\theta_{di}) \cdot l_s \cdot \omega_s}{2 \pi D} \quad (2.9)$$

Differentiating Eqn. (2.9)

$$\alpha_d = \ddot{\theta}_d = \cos^2(\theta_{di})\left[\frac{l_s \alpha_s}{2 \pi D} - 2\cos(\theta_{di}).\sin(\theta_{di})\left(\frac{l_s \omega_s}{2 \pi D}\right)^2\right] \quad \ldots \quad (2.10)$$

2.2.1.2 <u>Constant Ratio Steering Mechanism</u>

θ_{gin} : Angular rotation of input to gear box

ω_{gin} : Angular velocity of input to gear box

α_{gin} : Angular acceleration of input to gear box

R : Constant ratio

θ_d : Angular rotation of drop arm

ω_d : Angular velocity of drop arm

α_d : Angular acceleration of drop arm

Kinematic Equations are as follows:

$$\theta_d = R \cdot \theta_{gin} \quad (2.11)$$

$$\omega_d = R \cdot \omega_{gin} \quad (2.12)$$

$$\alpha_d = R \cdot \alpha_{gin} \quad (2.13)$$

2.2.1.3 Rack and Pinion Steering Mechanism

θ_{gin} : Angular rotation of input to gear box

ω_{gin} : Angular velocity of input to gear box

α_{gin} : Angular acceleration of input to gear box

D_p : Pitch diameter of the pinion

D_d : Displacement of the rack

V_d : Velocity of the rack

A_d : Acceleration of the rack

Kinematic equations are as follows:

$$D_d = \frac{D_p}{2} \cdot \theta_{gin} \tag{2.14}$$

$$V_d = \frac{D_p}{2} \cdot \omega_{gin} \tag{2.15}$$

$$A_d = \frac{D_p}{2} \cdot \alpha_{gin} \tag{2.16}$$

2.2.2 Description of the Subroutine Used for Kinematic Analysis of Steering Gear Box

Name of the subroutine : 'grbox'

Followings are the inputs to the subroutine:

$\underline{u}$gin : Unit vector along the input axis of gear box (pointing towards the gear box)

thtgin: Angular rotation of the input to the gear box

omggin: Angular velocity of the input to the gear box

alpgin: Angular acceleration of the input to the gear box

igbxtp: Type of gear box

igbxtp = 1 : Screw and nut type

igbxtp = 2 : Constant ratio type

igbxtp = 3 : Rack and pinion type

It screw and nut type gear box then following additional inputs are required

d : Length of the spindle arm

si : Initial nut displacement along the screw axis

rlead : Lead of the screw

For the constant ratio type gear box the ratio is required

$$\text{ratio} = \frac{\omega_{output}}{\omega_{input}}$$

For screw and nut type as well as constant ratio gear box gama 5 and gama 6 are required

gama 5 , gama 6 : Angle between Z axis and the projection, of the unit vector j along the axis of rotation of output shaft of the gear box, in YZ and ZX planes respectively.

If rack and pinion type gear box thenfollowing inputs are also required:

pd : Pitch diameter of the pinion

isign = + 1 (If rack is in the front of the pinion)

- 1 (If rack is at the back of the pinion)

Output of the subroutine is given below:

dspgbx : Output angular or linear (in case of rack and pinion) displacement of gear box

velgbx : Output angular or linear velocity of gear box

accgbx : Output angular or linear acceleration of gear box

uout : Unit vector along the axis of rotation of output shaft of the gear box

Algorithm for 'grbox' is given below

```
grbox →[Read (igbxtp)
        Case (igbxtp) of
        1  : [ { Screw and nut type }
               Read (d , si)
               Read (rlead)
               Read (gama5 , gama6 )
               Calculate uout
               Calculate h = nut displacement
                             from the orthogonal
                             configuration
                               (Using Eqn. 2.5)
               Calculate dspgbx (Using Eqn.2.8)
               Calculate omggbx (Using Eqn.2.9)
               Calculate alpgbx (Using Eqn.2.10)]
```

```
2 : [ {constant ratio type}
      Read (ratio)
      Read (gama 5 , gama6 )
      Calculate output
      dspgbx  = thtgin.ratio
      velgbx  = velgin.ratio
      accgbx  = accgin.ratio]
3 : [{ rack and pinion type }
      Read (pd)
      Read (isign)
      dspgbx = isign.thtgin.pd/2
      velgbx = isign.omggin.pd/2
      accgbx = isign.alpgin.pd/2]> grbox
```

2.3 Steering Linkage

As mentioned before, steering linkages consist of a combination of RSSR and SSR linkages. In the present analysis separate modules have been written for kinematic analysis of these two types of linkages, which are called repeatedly for the analysis of various steering linkages.

2.3.1 SSR Linkage

(Refer Fig. 1.6)

Though SSR is an open chain and not a mechanism in itself, it is used in combination with RSSR mechanism to constitute different steering linkages. It has three degrees of freedom - the coordinates of the first spherical joint.

2.3.1.1 Notations and Kinematic Equations

$\underline{a}_1$: Position vector of the point 'A' in the initial configuration

$\underline{a}$: Position vector of the point 'A' in the present configuration

$\dot{\underline{a}}$: Velocity vector of the point 'A'

$\ddot{\underline{a}}$: Acceleration vector of the point 'A'

$\underline{b}_o$: Position vector of the point 'B_o'

$\underline{b}_1$: Position vector of the point 'B' in the initial configuration

$\underline{b}$: Position vector of the point 'B' in the present configuration

$\dot{\underline{b}}$: Velocity vector of the point 'B'

$\ddot{\underline{b}}$: Acceleration vector of the point 'B'

β : Angular displacement of the output crank '$\overline{B\ B_o}$'

$\dot{\beta}$: Angular velocity of the output crank

$\ddot{\beta}$: Angular acceleration of the output crank

$\underline{u}_b$: Unit vector along the axis of rotation of output link.

Kinematic Equations for the SSR linkage have been given by Suh and Radcliffe [1]. These can be summarised as follows:

Given $\underline{b}_0$, $\underline{b}_1$, $\underline{u}_b$ and β , the position vector $\underline{b}$ of point B can be found out from equation

$$\underline{b} = \underline{b}_0 + [R_{\underline{u}_b}, \beta] (\underline{b}_1 - \underline{b}_0) \tag{2.17}$$

where the rotation matrix $[R_{\underline{u}_b}, \beta]$ is as follows [1] :

$$[R_{\underline{u}_b}, \beta] = \left[\begin{array}{c|c|c} u^2\, V(\beta) + C(\beta) & uv\, V(\beta) - wS(\beta) & uw\, V(\beta) + v\, S(\beta) \\ uv\, V(\beta) + w\, S(\beta) & v^2\, V(\beta) + C(\beta) & vw\, V(\beta) - u\, S(\beta) \\ uw\, V(\beta) - v\, S(\beta) & vw\, V(\beta) + uS(\beta) & w^2\, V(\beta) + C(\beta) \end{array}\right] \quad .. (2.13)$$

where

$$\underline{u}_b = (u, v, w)^T$$

$$V(\beta) = \text{vers}\ \beta = 1 - \cos\beta$$

$$S(\beta) = \sin\beta$$

$$C(\beta) = \cos\beta$$

$[R_{\underline{u}_b}, \beta]$ can be written in compact form given below

$$[R_{\underline{u}_b}, \beta] = - [P_{\underline{u}_b}] [P_{\underline{u}_b}] \cos\beta + [P_{\underline{u}_b}] \sin\beta + [Q_{\underline{u}_b}] \tag{2.19}$$

where

$$[P_{\underline{u}_b}] = \begin{bmatrix} 0 & -w & v \\ w & 0 & -u \\ -v & u & 0 \end{bmatrix}$$

$$[Q_{\underline{u}_b}] = \begin{bmatrix} u^2 & uv & uw \\ uv & v^2 & vw \\ uw & vw & w^2 \end{bmatrix}$$

$$-[P_{\underline{u}_b}][P_{\underline{u}_b}] = [I - Q_{\underline{u}_b}]$$

where

I : Identity matrix

Another constraint to be satisfied is constant length of link AB. This can be expressed as follows:

$$(\underline{a} - \underline{b})^T (\underline{a} - \underline{b}) = (\underline{a}_1 - \underline{b}_1)^T (\underline{a}_1 - \underline{b}_1) \quad (2.20)$$

Substituting $\underline{b}$ from Eqn. 2.17 in Eqn. 2.20 we get

$$E \cos \beta + F \sin \beta + G = 0 \quad (2.21)$$

where

$$E = (\underline{a} - \underline{b}_0)^T [I - Q_{\underline{u}_b}] (\underline{b}_1 - \underline{b}_0) \quad (2.22)$$

$$F = (\underline{a} - \underline{b}_0)^T [P_{\underline{u}_b}] (\underline{b}_1 - \underline{b}_0) \quad (2.23)$$

$$G = (\underline{a} - \underline{b}_0)^T [Q_{\underline{u}_b}] (\underline{b}_1 - \underline{b}_0) + \frac{1}{2} [(\underline{a}_1 - \underline{b}_1)^T (\underline{a}_1 - \underline{b}_1) - (\underline{a} - \underline{b}_0)^T (\underline{a} - \underline{b}_0) - (\underline{b}_1 - \underline{b}_0)^T (\underline{b}_1 - \underline{b}_0)] \quad (2.24)$$

Eqn. (2.20) can be solved to give

$$\beta = 2 \tan^{-1} \left[\frac{-F \pm \sqrt{E^2 + F^2 - G^2}}{G - E} \right] \quad (2.25)$$

$\dot{\underline{b}}$ and $\ddot{\underline{b}}$ can be calculated from equations given below:

$$\dot{\underline{b}} = \dot{\beta}\,[P_{\underline{u}_b}]\,(\underline{b} - \underline{b}_0) \qquad (2.26)$$

$$\ddot{\underline{b}} = [\,\ddot{\beta}\,[P_{\underline{u}_b}] + \dot{\beta}^2\,[P_{\underline{u}_b}]\,[P_{\underline{u}_b}]]\,(\underline{b} - \underline{b}_0) \qquad \ldots \quad (2.27)$$

The angular velocity and acceleration of the output crank are

$$\dot{\beta} = \frac{(\underline{a} - \underline{b})^T\,\dot{\underline{a}}}{(\underline{a} - \underline{b})\,[P_{\underline{u}_b}]\,(\underline{b} - \underline{b}_0)} \qquad (2.28)$$

$$\ddot{\beta} = \frac{(\dot{a}-\dot{b})^T\,(\dot{a}-\dot{b}) + (\underline{a}-\underline{b})^T\,(\ddot{\underline{a}}-\beta^2\,[P_{\underline{u}_b}][P_{\underline{u}_b}]\,(\underline{b}-\underline{b}_0))}{(\underline{a}-\underline{b})^T\,[P_{\underline{u}_b}]\,(\underline{b}-\underline{b}_0)} \qquad \ldots \quad (2.29)$$

2.3.1.2 Description of Subroutine Used for Kinematic Analysis of SSR Linkage

Name of the subroutine : ssr

Followings are the input variables for the subroutine

Ub : Unit vector along axis of rotation of output link

a1 , b1 , bo: Position vectors of the joints in initial configuration

a : Position vector of the input point in present configuration

adot : Velocity vector of the input point in present configuration

addot : Acceleration vector of the input point in present configuration

Output of the subroutine is given below:

mobile = . True . if the SSR linkage can be assembled for the given value of $\underline{a}$ otherwise . False .

If (mobile = . True .) then following additional output is also obtained

beta : Angular displacement of output link

btdot : Angular velocity of output link

btddot : Angular acceleration of output link

$\underline{b}$: Position vector of point B, the joint between coupler and output link

$\underline{\dot{b}}$dot : Velocity vector of point B

$\underline{b}$ddot : Acceleration vector of point B

Algorithm for subroutine ssr is given below:

ssr →[Assign the values of all the elements of matrices [p],[q] and [t] for unit vector $\underline{U}_b$

btddot = fbddot ($\underline{a}_1$, $\underline{a}$, $\underline{b}_1$, $\underline{b}_0$, a$\underline{d}$ot , a$\underline{d}$dot , [p], [q] , [t])

fbddot →[btdot = fbdot ($\underline{a}_1$, $\underline{a}$, $\underline{b}_1$, $\underline{b}_0$, a$\underline{d}$ot , [p], [q], [t])

fbdot →[beta = fbeta ($\underline{a}_1$, $\underline{a}$, $\underline{b}_1$, $\underline{b}_0$, [p], [q],[t])

fbeta →[mobile = . True .

calculate e, f, g (Using Eqns. (2.22), (2.23), (2.24))

```
            det = e^2 + f^2 - g^2
            if (det < 0.0) then
               mobile = . false .
            else
               calculate beta1 , beta2
                   (Using Eqn. 2.25)
               fbeta  =  min (beta1 , beta2 )
            endif  ] -> fbeta
       if (mobile) then
           calculate b (Using Eqn. 2.17)
           fbdot = β̇  (Using Eqn. 2.23)
       endif ]-> fbdot
  if (mobile) then
       calculate bdot (Using Eqn. 2.26)
       fbddot = β̈ (Using Eqn. 2.29)
       Calculate bddot(Using Eqn. 2.27)
  endif]-> fbddot]-> ssr
```

2.3.2 RSSR Linkage

A RSSR linkage can be considered to be made up of a SSR linkage with an additional link, the rotation of which specifies the coordinates of input point of SSR linkage. It has only one degree of freedom - the amount of rotation of input link.

2.3.2.1 Notations and Kinematic Equations (Refer Fig. 1.7)

Except the notations given below, remaining notations are same as used in Section 2.3.1.1 for the analysis of SSR linkage.

$\underline{U}_a$: Unit vector along the input axis

$\underline{a}_0$: Position vector of point A_0

α : Angular displacement of input link

$\dot{\alpha}$: Angular velocity of input link.

$\ddot{\alpha}$: Angular acceleration of input link

Kinematic equations derived for SSR linkage can be used for RSSR linkage also. For RSSR linkage displacement, velocity and acceleration vectors for point A can be calculated using following equations:

$$\underline{a} = \underline{a}_0 - [R_{\underline{u}_a}, \alpha] (\underline{a}_1 - \underline{a}_0) \tag{2.30}$$

$$\dot{\underline{a}} = \dot{\alpha} [P_{\underline{u}_a}] (\underline{a} - \underline{a}_0) \tag{2.31}$$

$$\ddot{\underline{a}} = [\ddot{\alpha} [P_{\underline{u}_a}] + \dot{\alpha}^2 [P_{\underline{u}_a}] [P_{\underline{u}_a}]] (\underline{a} - \underline{a}_0) \tag{2.32}$$

Here $[R_{\underline{u}_a}, \alpha]$ and $[P_{\underline{u}_a}]$ are same as described in Section 2.3.1.

2.3.2.2 Description of the Subroutine Used for the Kinematic Analysis of RSSR Linkage

Name of the subroutine : rssr

Following is the input to the subroutine:

$\underline{\text{Ua}}$	: Unit vector along the input axis
$\underline{\text{Ub}}$	: Unit vector along the output axis
$\underline{\text{a}}0$, $\underline{\text{a}}1$, $\underline{\text{b}}1$, $\underline{\text{b}}0$	: Position vectors of the joints in the initial configuration
alpha	: Angular displacement of the input link from the initial configuration
aldot	: Angular velocity of the input link
alddot	: Angular acceleration of the input link

Output of the subroutine is given below:

mobile	: . True . if the RSSR linkage can be assembled for the given value of 'alpha' otherwise . false .
beta	: Angular displacement of the output link from the initial configuration
btdot	: Angular velocity of the output link
btddot	: Angular acceleration of the output link
$\underline{\text{a}}$	: Position vector of point A-the joint between the coupler and the input link
a$\underline{\text{d}}$ot	: Velocity vector of point A
a$\underline{\text{d}}$dot	: Acceleration vector of point A
$\underline{\text{b}}$	: Position vector of point B the joint between the coupler and output link
b$\underline{\text{d}}$ot	: Velocity vector of point B
b$\underline{\text{d}}$dot	: Acceleration vector of point B

Algorithm for subroutine rssr

rssr → [Assign the values to all the elements of matrices [p], [q] and [t] for $\underline{Ua}$.

Calculate $\underline{a}$ (Using Eqn. 2.30)

Calculate $\underline{adot}$(Using Eqn. 2.31)

Calculate $\underline{addot}$(Using Eqn. 2.32)

Call SSR ($\underline{Ub}$, $\underline{a1}$, $\underline{a}$, $\underline{b1}$, $\underline{b0}$, $\underline{adot}$, $\underline{addot}$)] → rssr

2.4 Steering Error

Steering error as explained in Section 1.3 can be measured in three different ways. (Refer Fig. 2.5)

1. Perpendicular distance from the axis of rear wheels to the point G, where axes of rotation of two front wheels intersect. (Distance Error).
2. Area of triangle EFG, in Fig. 2.5, can be taken as a measure of the steering error. (Area Error)
3. Steering error can also be measured as the difference between actual and ideal angles through which the left wheel is turned, i.e. angle EA_0F in Fig. 2.5. (Angle Error).

2.4.1 Notations and Kinematic Equations

$\underline{u}_{rt}$: Unit vector along the axis of right tire, when the vehicle is moving on a straight path

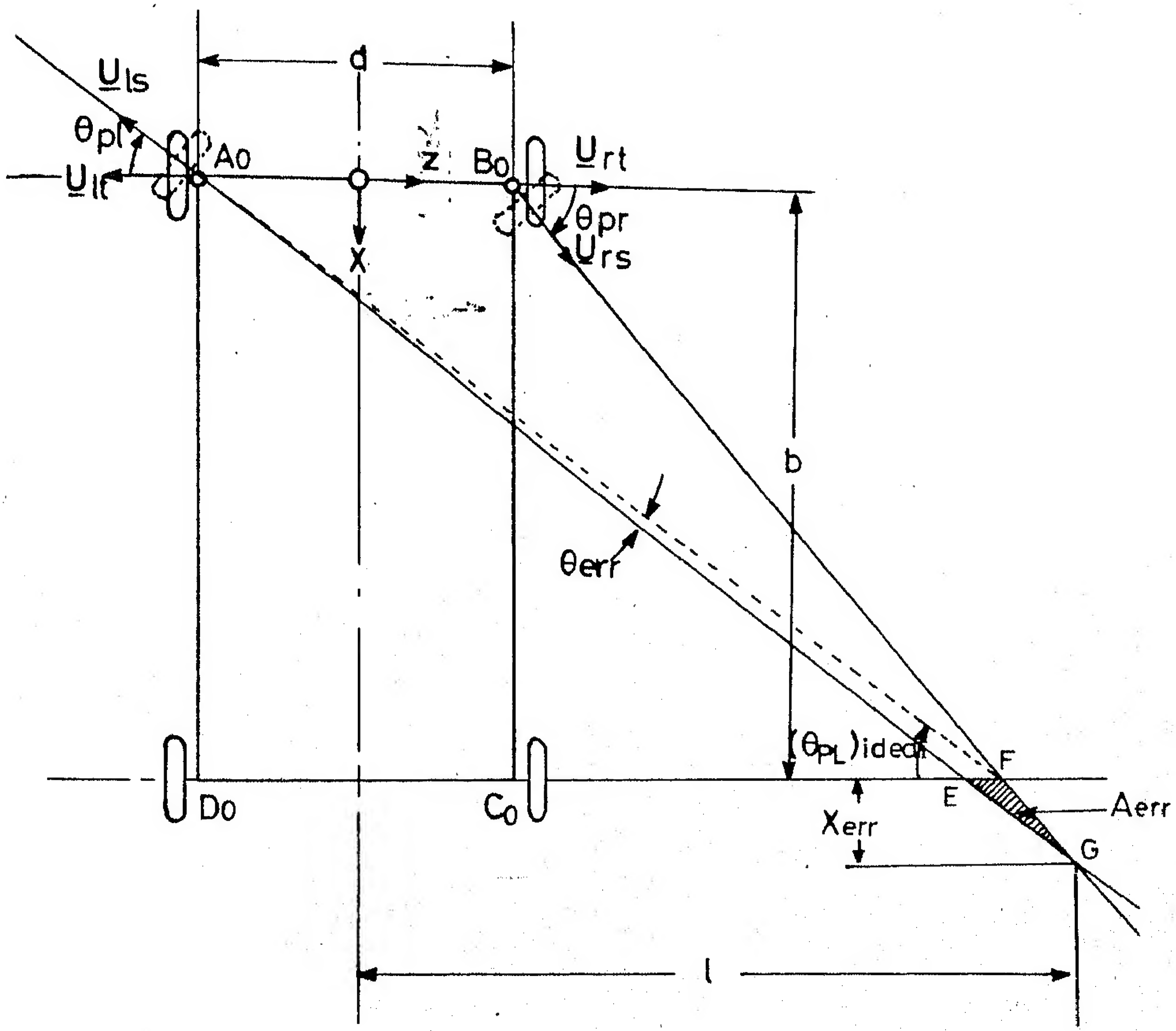


Fig.2.5 Steering error.

$\underline{u}_{lt}$: Unit vector along the axis of left tire, when the vehicle is moving on a straight path

$\underline{u}_{rs}$: Unit vector along the axis of right tire in a steered configuration

$\underline{u}_{ls}$: Unit vector along the axis of left tire in a steered configuration

$\underline{u}_{rk}$: Unit vector along right kingpin

$\underline{u}_{lk}$: Unit vector along left kingpin

θ_r : Angle through which right tire is rotated about the kingpin

θ_l : Angle through which left tire is rotated about the kingpin

θ_c : Camber angle

θ_{pr} : Angle between Z axis and the projection of vector $\underline{u}_{rs}$ in XZ plane

θ_{pl} : Angle between Z axis and the projection of vector $\underline{u}_{ls}$ in XZ plane

b : Wheel base

d : Distance between the two king pins

l : As shown in Fig. 2.5

X_{err} : Distance error

A_{err} : Area error

θ_{err} : Angle error

Kinematic equations for three types of steering error are given below:

For a vehicle going on a straight path unit vectors along the axis of rotation are:

for right tire

$$\underline{u}_{rt} = \begin{Bmatrix} 0.0 \\ \sin \theta_c \\ \cos \theta_c \end{Bmatrix} \tag{2.34}$$

for left tire

$$\underline{u}_{lt} = \begin{Bmatrix} 0.0 \\ \sin \theta_c \\ -\cos \theta_c \end{Bmatrix} \tag{2.35}$$

Unit vectors along the axes of tires after the vehicle has been steered can be calculated from the following equations:

$$\underline{u}_{rs} = [R_{\underline{u}_{rk}}, \theta_r] \cdot \underline{u}_{rt} \tag{2.35}$$

$$\underline{u}_{ls} = [R_{\underline{u}_{lk}}, \theta_l] \cdot \underline{u}_{lt} \tag{2.36}$$

Tangent of angles made by projections of $\underline{u}_{rs}$ and $\underline{u}_{ls}$ in ZX plane, with Z axis are

$$\tan (\theta_{pr}) = \frac{(\underline{u}_{rs})_x}{(\underline{u}_{rs})_z} \tag{2.37}$$

$$\tan (\theta_{pl}) = \frac{(\underline{u}_{ls})_x}{(\underline{u}_{ls})_z} \tag{2.38}$$

As shown in Fig. 2.5

$$\tan(\theta_{pr}) = \frac{b + X_{err}}{1 - d/2} \qquad (2.39)$$

$$\tan(\theta_{pl}) = \frac{b + X_{err}}{1 + d/2} \qquad (2.40)$$

Using Eqns. (2.35) - (2.38)

$$X_{err} = d\left(\frac{\tan\theta_{pl} \cdot \tan\theta_{pr}}{\tan\theta_{pr} - \tan\theta_{pl}}\right) - b \quad \ldots \quad (2.41)$$

Area error (Refer Fig. 2.5)

$$A_{err} = \frac{(FD_O - ED_O)\, X_{err}}{2}$$

$$= \frac{X_{err}}{2}\left(b\left(\frac{\tan\theta_{pl} - \tan\theta_{pr}}{\tan\theta_{pl} \cdot \tan\theta_{pr}}\right) + d\right) \quad \ldots \quad (2.42)$$

Angle error (Refer Fig. 2.5)

$$\theta_{err} = (\theta_{pl})_{actual} - (\theta_{pl})_{ideal}$$

$$= \angle E A_O B_O - \angle F A_O B_O$$

$$= \angle E A_O B_O - \tan^{-1}\left(\frac{b \tan\theta_{pr}}{b+d \tan\theta_{pr}}\right) \quad \ldots\ldots \quad (2.43)$$

2.4.2 <u>Description of the Subroutine Used for Calculating Steering Error</u>:

Name of subroutine : strerr

Following variables are required as input to the subroutine 'strerr':

urkp	:	Unit vector along right king pin
ulkp	:	Unit vector along left king pin
thtrgt	:	Angle through which the right tire is rotated about the king pin
thtlft	:	Angle through which the left tire is rotated about the king pin
cmbr	:	Camber angle
dkpin	:	Distance between two king pins

Output of the subroutine is given below:

xerr	:	Distance error
arerr	:	Area error
angerr	:	Angle error

Algorithm for subroutine strerr is as follows:

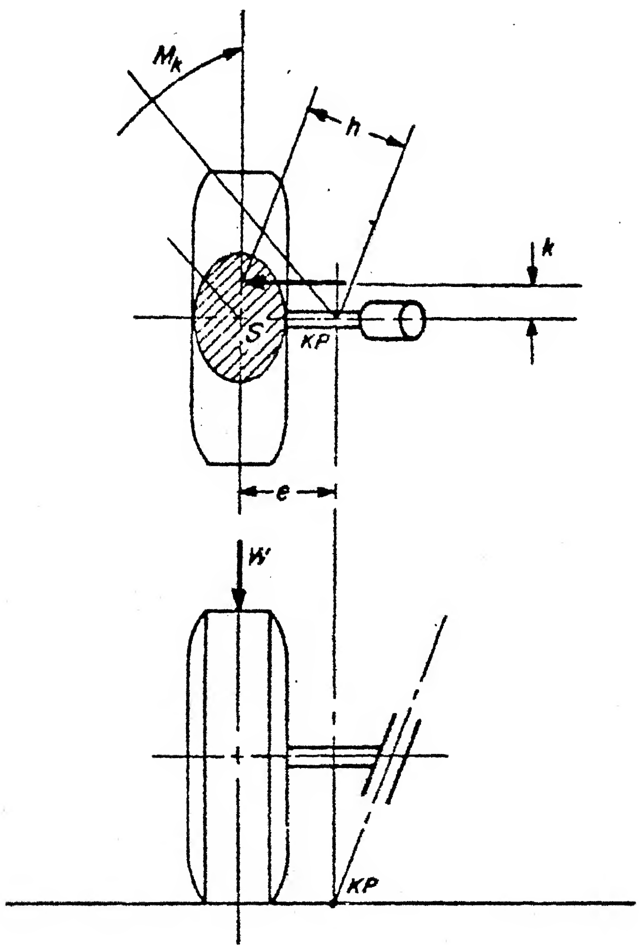
```
strerr → Read (cmbr)
         Calculate unit vectors along the axis
         of two tires (Using Eqns. (2.33)(2.34))
         Calculate unit vectors along the axis
         of two tires in the steered configu-
         ration      (Using Eqns. (2.35) (2.36))
         Read  wb
         if (moving on a straight path) then
               xerr = 0.0
              arerr = 0.0
             angerr = 0.0
         else
```

Chapter 3

FORCE ANALYSIS OF THE CONSTITUENT MECHANISMS OF A STEERING SYSTEM

3.1 Moment Required to Steer a Stationary Vehicle

When a wheel is turned around the king pin, the hypothetical intersection of the king pin with the ground acts as the center of rotation. (Point KP in Fig. 3.1). Turning of the wheel around the center of rotation will be pure slipping if king pin offset e, is zero. In ideal case of line contact between road and the tire, for value of king pin offset between zero and half of the tire width b_t , turning of the wheel will be a combination of rolling and slipping and for $e > b_t/2$, the turning of the wheel will be pure rolling. The contact region between the tire and the road as shown in Fig. 3.1 is not a line and therefore even for the case of $e > b_t/2$, turning of the wheel about the center of rotation KP is a combination of rolling and slipping. Mathematical equations for calculating the moment required to turn a stationary wheel will be highly complicated. Taborek [4] derived empirical equations for calculating it. These are summarised below.



Fig·3·1 Turning moment & turning center of a steered wheel·

3.1.1 Notations and Equations

b_t : Nominal tire width

e : King pin offset

h_t : Effective torque arm

k_t : Polar radius of gyration of contact region between the tire and the road

M_k : Moment required to turn the stationary wheel

w_t : Load on the wheel

μ : Effective friction coefficient.

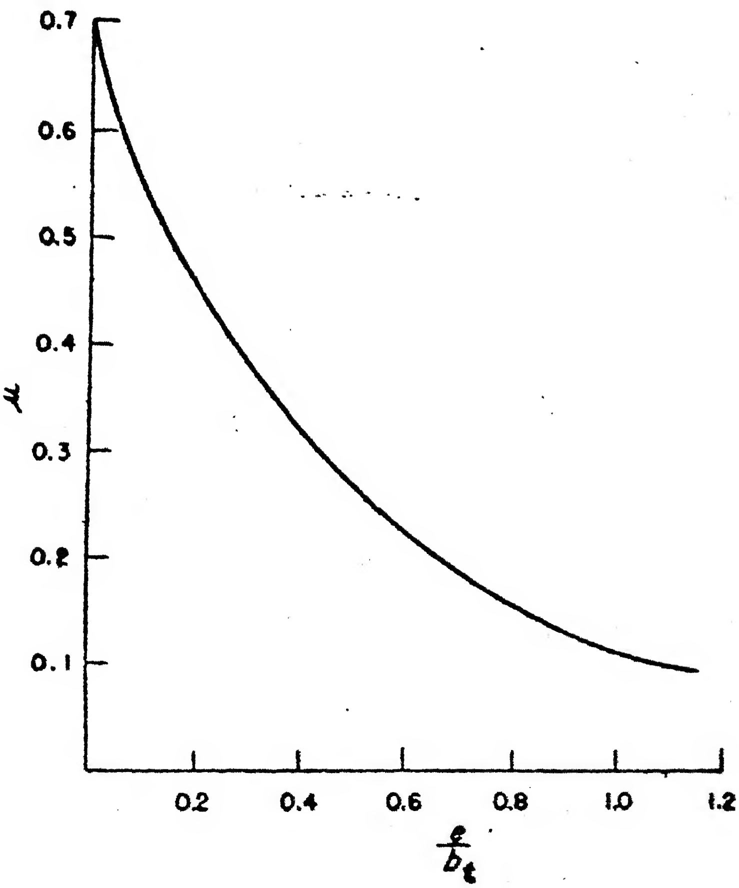
The torque required to turn the wheel is the moment of the integral friction forces about the center of rotation KP. The integral of frictional forces is assumed to be acting at a distance equal to the polar radius of gyration of the contact region. The effect torque arm becomes

$$h_t = \sqrt{e^2 + k_t^2} \tag{3.1}$$

Therefore the torque required to turn the wheel is

$$M_k = \mu \, w_t h \tag{3.2}$$

Here the effective friction coefficient is not a constant but a function of king pin offset e and nominal tire width b_t. On concrete roads (static friction coefficient $\mu_s = 0.70$), approximate values of μ are given in Fig. 3.2 as a function of e/b_t.



Fig·3·2 Effective friction coefficient between a steered tire and road·

For the normal load conditions, the contact region between tire and road can be approximated to a circle of diameter b_t , the nominal tire width. Therefore

$$k_t^2 = \frac{I_O}{A} = \frac{b_t^2}{8} \tag{3.3}$$

where

A_O : Area of the contact region

I_O : Polar moment of inertia of the area of the contact region.

Because of the king pin inclination, the whole axle must be lifted as the wheels are turned. Therefore an additional moment is required to steer the stationary vehicle. For the usual king pin inclination of about 6°, this additional moment is relatively small compared to the frictional component of the torque.

3.2 Steering Linkage

For force analysis of steering linkages, like the kinematic analysis, two separate subroutines have been written for force analysis of SSR and RSSR linkages, which are repeatedly called to analyse the steering linkage. As the angular and linear acceleration of various links is small, the magnitude of inertial force, compared to the external forces and internal forces, is

negligible. In the present analysis the inertial force has been neglected and only static force analysis has been done.

3.2.1 SSR Linkage
(Refer Fig. 3.3)

For the force analysis of SSR linkage, it is assumed that the external forces acting on the two links and the axial moment acting on the crank are known. The system of internal forces f_i and force required at the open end to balance the axial moment on the crank are to be calculated.

3.2.1.1 Notations and Equations

$\underline{a}$: Position vector of the open end

$\underline{b}, \underline{b}_0$: Position vectors of the joints

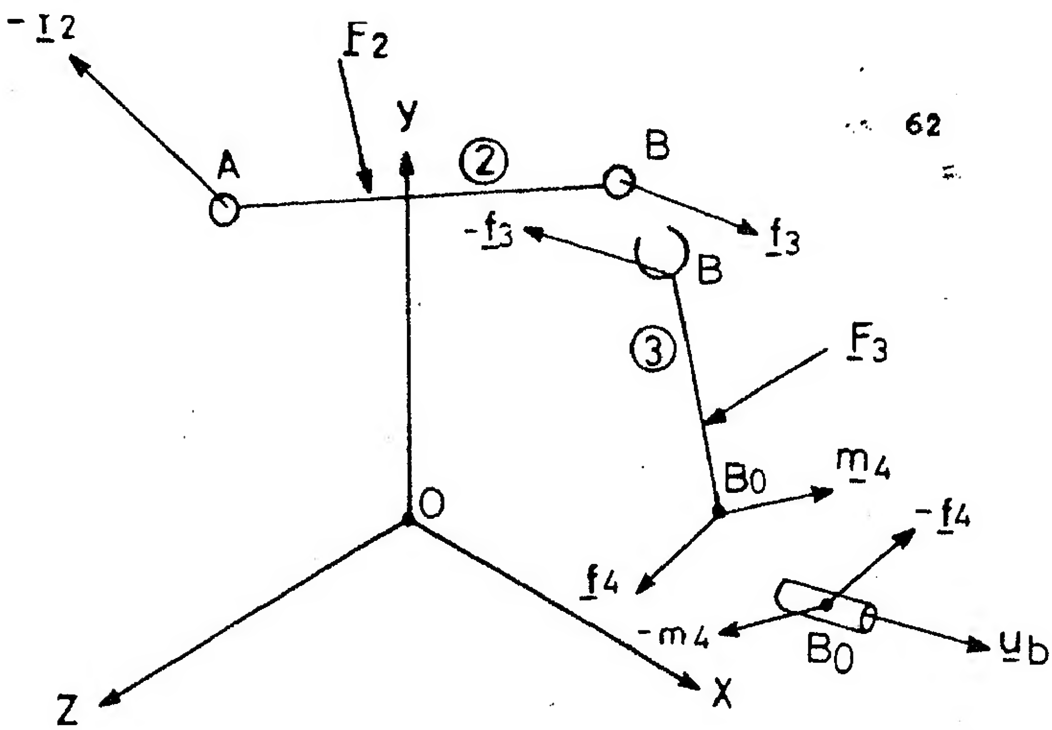
$\underline{f}_i$: Internal force applied by i^{th} link on $(i-1)^{th}$ link

$\underline{F}_i$: External force acting on i^{th} link

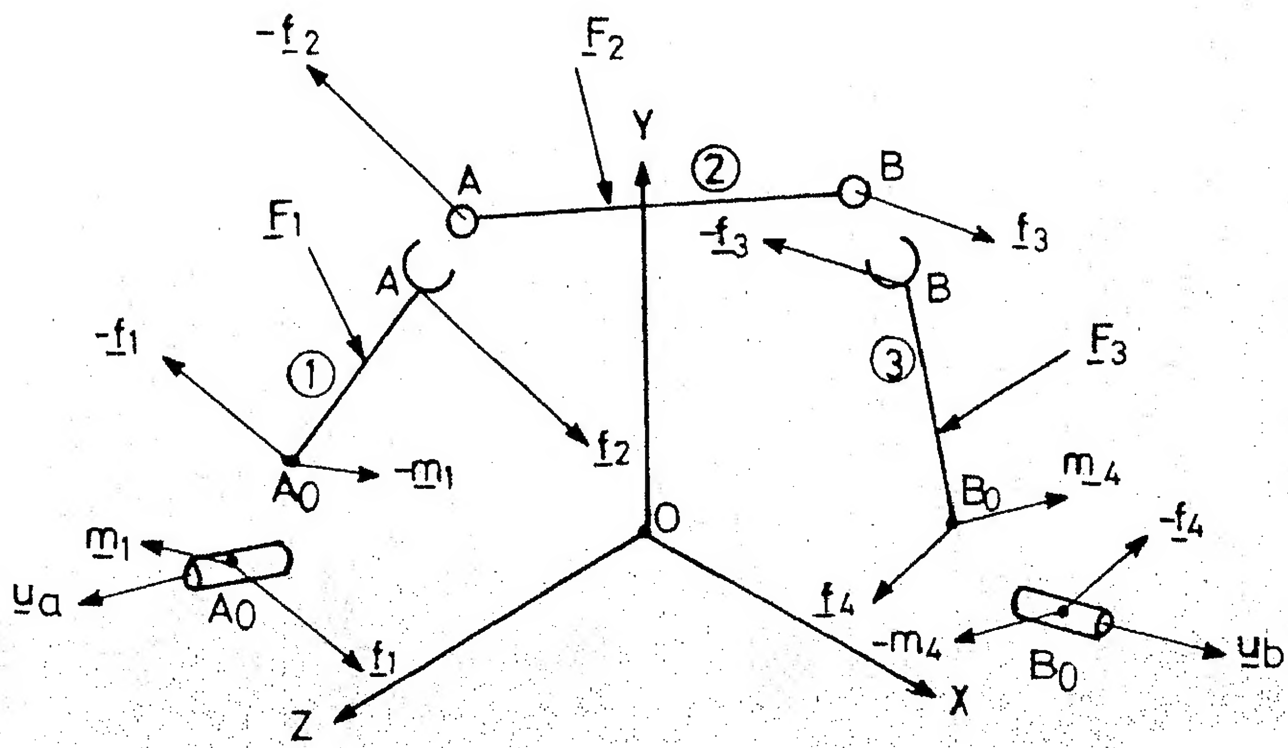
k_i : $\dfrac{\text{Distance from } (i-1)^{th} \text{ link to the point of application of external force on } i^{th} \text{ link}}{\text{length of the } i^{th} \text{ link}}$

m_{4a} : External axial moment on the crank

$\underline{m}_4$: Total moment applied on link $B\,B_0$ at the joint B_0 (i.e. vector sum of external axial moment and moment applied by the fixed link)



Fig·3·3 Free body diagram of S-S-R linkage·



Fig·3·4 Free body diagram of R-S-S-R linkage·

$\underline{u}_b$: Unit vector along the axis of rotation of crank.

Fig. 3.3 gives the free body diagram of the two moving links of a SSR linkage. Writing the force and moment balance equations for the two links separately we get the equations for unknown forces and moments.

For link AB, force and moment balance equations are

$$-\underline{f}_2 + \underline{F}_2 + \underline{f}_3 = 0 \qquad (3.4)$$

$$(\underline{b} - \underline{a}) \times \underline{f}_3 + k_2 (\underline{b} - \underline{a}) \times \underline{F}_2 = 0 \qquad (3.5)$$

where

$$k_2 = \frac{\overline{AE_2}}{\overline{AB}}$$

Similarly for link B B_O , force and moment balance equations are

$$-\underline{f}_3 + \underline{F}_3 + \underline{f}_4 = 0 \qquad (3.6)$$

$$(\underline{b}_O - \underline{b}) \times \underline{f}_4 + k_3 (\underline{b}_O - \underline{b}) \times \underline{F}_3 + \underline{m}_4 = 0 \qquad \ldots (3.7)$$

One more equation can be obtained knowing the axial component of $\underline{m}_4$

$$\underline{m}_4 \cdot \underline{u}_b = m_{4a} \qquad (3.3)$$

Here $\underline{f}_2$, $\underline{f}_3$, $\underline{f}_4$ and $\underline{m}_4$ are unknown. In equation 3.5, for both the terms on the left hand side the component along vector $(\underline{b} - \underline{a})$ is zero, therefore even though a

vector equation, it gives only two independent algebric equations. Any two out of the three equations can be selected. But if one of the principal axis is along the link then component along that axis should be avoided. For this, the component of equation(3.2)along the principal axis along which vector ($\underline{b}$ - $\underline{a}$) has the largest component is rejected.

Equations (3.4) to (3.8) can be combined to get a system of twelve algebric equations in terms of three components of $\underline{f}_2$, $\underline{f}_3$, $\underline{f}_4$ and $\underline{m}_4$ each which can be solved using Gauss-elimination method. Matrix representation of the system is given on the next page. In this

$$\underline{r}^{ba} = \underline{b} - \underline{a}$$

$$\underline{r}^{b_0 b} = \underline{b}_0 - \underline{b}$$

3.2.2 RSSR Linkage

Fig 3.4 shows the free body diagram of three moving links of a RSSR linkage. Once SSR part of the RSSR linkage has been analysed for the input crank of RSSR linkage, force $\underline{f}_2$, applied by the open end of SSR linkage, is taken as a known force and the external axial moment and the internal force and moment at the joint between crank and fixed link are calculated.

$$
\begin{bmatrix}
1 & 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 1 & 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & -r_z^{ba} & r_y^{ba} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & r_z^{ba} & 0 & -r_x^{ba} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & -r_y^{ba} & r_x^{ba} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 1 & 0 & 0 & -1 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 1 & 0 & 0 & -1 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 & -1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & -r_z^{b_0 b} & r_y^{b_0 b} & 1 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & \underline{r}_z^{b_0 b} & 0 & -r_x^{b_0 b} & 0 & 1 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & -r_y^{b_0 b} & r_x^{b_0 b} & 0 & 0 & 0 & 1 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & u_{bx} & u_{by} & u_{bz}
\end{bmatrix}
\begin{bmatrix}
f_{2x} \\ f_{2y} \\ f_{2z} \\ f_{3x} \\ f_{3y} \\ f_{3z} \\ f_{4x} \\ f_{4y} \\ f_{4z} \\ m_{4x} \\ m_{4y} \\ m_{4z}
\end{bmatrix}
=
\begin{bmatrix}
F_{2x} \\ F_{2y} \\ F_{2z} \\
\left.\begin{matrix} -k_2(\underline{r}^{ba} \times \underline{F}_2)_x \\ -k_2(\underline{r}^{ba} \times \underline{F}_2)_y \\ -k_2(\underline{r}^{ba} \times \underline{F}_2)_z \end{matrix}\right\} \text{Only two from these three} \\
F_{3x} \\ F_{3y} \\ F_{3z} \\
-k_3(\underline{r}^{b_0 b} \times \underline{F}_3)_x \\ -k_3(\underline{r}^{b_0 b} \times \underline{F}_3)_y \\ -k_3(\underline{r}^{b_0 b} \times \underline{F}_3)_z \\
m_{4a}
\end{bmatrix}
$$

3.2.2.1 Notations and Equations

In addition to the notations used in Section 3.2.1.1 for SSR linkage, following notations are also used for RSSR linkage.

m_{1a} : External axial moment applied on the input crank

$\underline{m}_1$: Total moment applied on link A A_O at the joint A_O. (i.e. the vector sum of external axial moment m_{1a} and moment applied by the fixed link)

$\underline{u}_a$: Unit vector along the axis of rotation of input crank.

Force and moment equations for input crank A A_O are

$$- \underline{f}_1 + \underline{F}_1 + \underline{f}_2 = 0$$

$$- \underline{m}_1 + k_1 (\underline{a} - \underline{a}_O) \times \underline{F}_2 + (\underline{a} - \underline{a}_O) \times \underline{f}_2 = 0$$

i.e.

$$\underline{f}_1 = \underline{F}_1 + \underline{f}_2 \tag{3.9}$$

$$\underline{m}_1 = k_1 (\underline{a} - \underline{a}_O) \times \underline{F}_2 + (\underline{a} - \underline{a}_O) \times \underline{f}_2 \tag{3.10}$$

External axial moment on link A A_O can be calculated from the equation

$$m_{1a} = \underline{m}_1 \cdot \underline{u}_a \tag{3.11}$$

3.3 Steering Effort

In the part of steering system from steering wheel to the Pitman arm, mechanical energy is transmitted through universal joints and steering gear box. Since there are no bifurcations for energy to get distributed in different channels, equation for steering effort can be obtained using the principle of energy conservation.

3.3.1 Notations and Equations

d_s : Diameter of steering wheel

F_s : Steering effort (i.e. hand force applied at steering wheel)

M_{apm} : Axial moment required to turn the Pitman arm

ω_d : Angular velocity of Pitman arm

ω_{st} : Angular velocity of steering wheel

η_g : Efficiency of gear box

Applying energy conservation principle for energy available at steering wheel and that at gear box output

$$\omega_{st} \cdot F_s \cdot \frac{d_s}{2} = \frac{\omega_d \cdot M_{apm}}{\eta_g}$$

$$F_s = \frac{2\omega_d \, M_{apm}}{\omega_{st} \cdot d_s \cdot \eta_g} \qquad (3.12)$$

Here the efficiencies of universal joints are assumed to be unity.

With steering effort calculations, all the forces and moments in the system are known. In the next Chapter, a typical steering system - one used in trucks - is analyzed completely using the programme. Coordinate frame and other conventions used in the programme are also clarified in the next Chapter.

Chapter 4

PROGRAM STRUCTURE

4.1 Block Diagram

Fig. 4.1 shows the flow chart of the program developed for the analysis of steering systems. In the figure various stages of the analysis have been shown inside rectangular boxes, whereas input required at various stages and the result obtained have been encircled. Options available at various stages of the analysis are at the bottom of the rectangular boxes. Output variables of final interest are position, velocity and acceleration vectors of spherical joints, angular displacement, velocity and acceleration of two wheels, steering error, internal forces and moments between various links and steering effort.

4.2 General Conventions Used in the Program

Assumptions about the type of input and output variables and units used are given below:

1. Except angles which are easier to visualize in degrees, all the input variables are to be specified in SI units. Angular velocities and

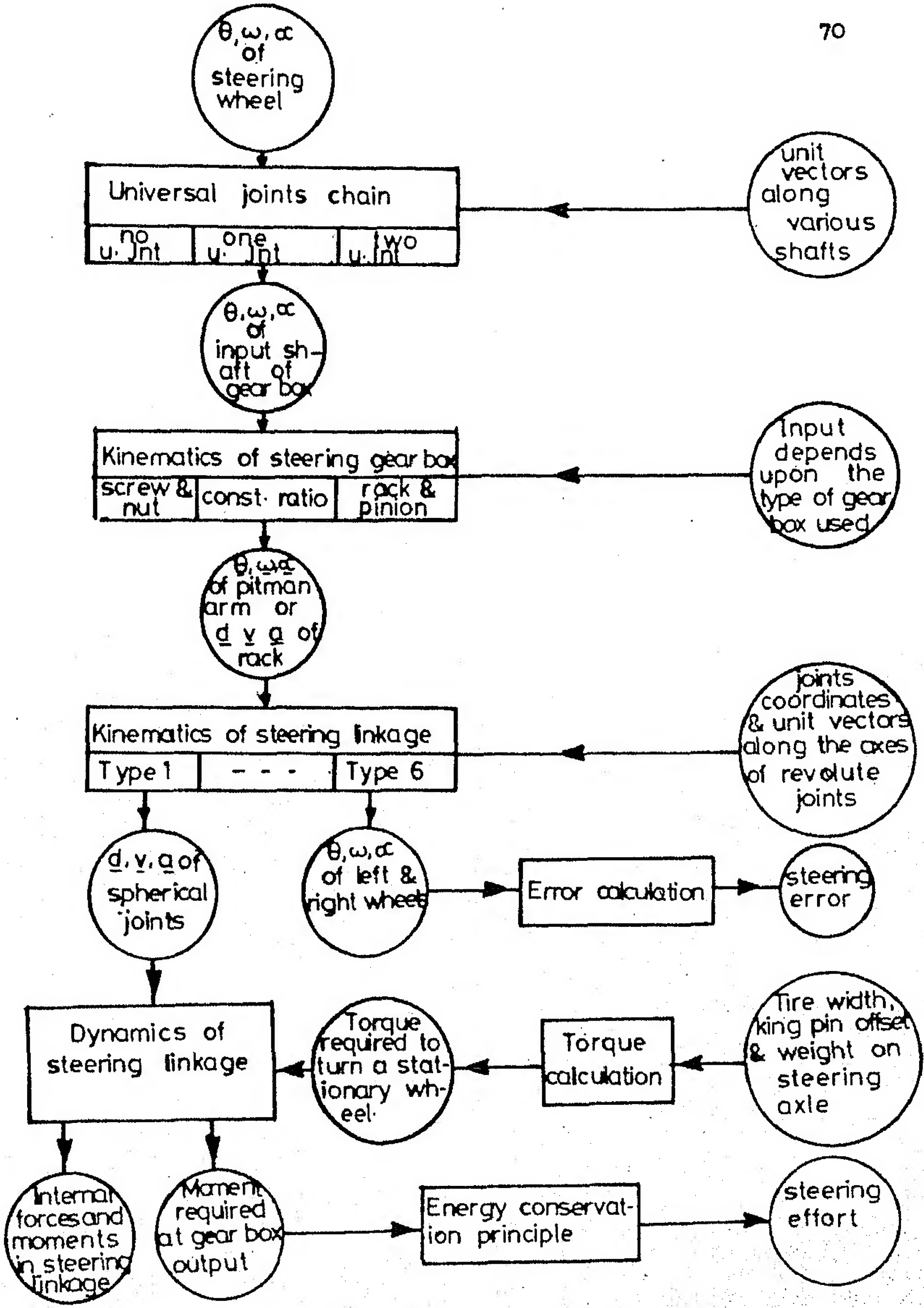


Fig. 4.1 Block diagram of the programme.

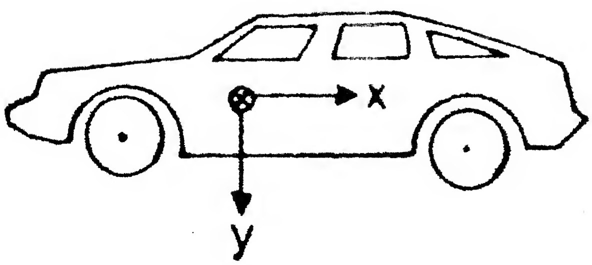
angular accelerations are still to be given in rad/sec and rad/sec^2 respectively.

2. Except count (e.g. number of universal joints) and type (e.g type of steering gear box) specifications, all the variables are assumed to be real numbers.

3. All the output variables are in SI units. For angles values in paranthesis are corresponding values in degrees.

4.3 Coordinate Frame

The coordinate frame used for all calculations, is shown in Fig. 4.2. Sitting at the driver's position, for the coordinate frame used, X axis goes from the front to the rear end of the vehicle, Y axis is vertically down and Z axis is directed side-way towards the right. As the magnitude and direction of the vectors along the links and along the axes of rotation are not affected by the position of the origin of the coordinate frame, it can be fixed at any convenient point on the vehicle. But the sense and the magnitude of the caster, the camber and the king pin inclination are assumed to be according to the general conventions for these, therefore the directions of principal axes should match with the coordinate frame shown in Fig. 4.2.



Z Axis is directed into the page.

Fig. 4.2 Coordinate frame.

4.4 Analysis of the Steering System of a Truck

Though even for trucks more than one type of steering systems are being used, one of the steering systems, very commonly used in trucks was shown in Fig. 1.3. It has two universal joints between steering wheel and steering gear box, screw and nut type of steering gear box and the spindle lever steering linkage between steering gear box and two steered wheels. Analysis of this steering system, using the developed program has been done. Except the comments in curly brackets, which are given here for clarification, all the comments printed on the screen by the computer, input variables typed by the user and the output of the program are given exactly the way these come on the screen. Input variables are assumed to be format free. These are underlined in the analysis given below.

Feed in the number of universal joints
2
Feed in the values of angular rotation, angular velocity and angular acceleration of steering wheel
200.0 1.0 0.0
Feed in the values of gama1, gama2, gama3 and gama4 in the figure shown {Fig. 2.3 displayed}
67.0 90.0 86.0 84.0
Feed in the coordinates of points P and Q

0.795 -0.55 0.695

-0.367 -0.124 0.695

For the first universal joint feed in the value of angle between common perpendicular to two shafts and the line passing through the points where input yoke is attached to the cross in the starting configuration

0.0

For the intermediate shaft

angular rotation = 0.3359 e + 01 (0.1925 e + 03)

angular velocity = 0.6555 e + 00

angular acceleration = 0.2873 e + 00

For the second universal joint feed in the value of angle between common perpendicular to the two shafts and the line passing through the points where input yoke is attached to the cross in the starting configuration

0.0

Following is the output of the universal-joints-chain

angular rotation = 0.3130 e + 01 (0. 1822 e + 03)

angular velocity = 0.1230 e + 00

angular acceleration= 0.3658 e - 01

Specify the type of gear box

type 1 if screw and nut type

type 2 if constant ratio type

type 3 if rack and pinion type

1

Feed in the length D and initial nut displacement SI as shown in the figure above { Fig. 2.4 displayed }

<u>0.06</u> <u>0.006</u>

Feed in the lead of the screw

<u>0.011</u>

In the figure shown, vector AB is along the axis of rotation of Pitman arm and it should be directed such that for input angular velocity directed into the gear box, angular velocity output from the gear box should be along AB{ Figure displayed}

Feed in the value of gama5 and gama6

<u>0.0</u> <u>6.0</u>

Output of the gear box

angular rotation = 0.4995 e + 00 (0.2862 e + 02)^

angular velocity = 0.2447 e - 02

angular acceleration = 0.1715 e - 02

in the direction u = 0.1045 i 0.000 j 0.9945 k

Specify the type of steering linkage from the linkages shown {Figs. 1.8 to 1.13 displayed.}

<u>1</u>

{Fig. 1.8 displayed}

Feed in the coordinates of points AO, A11, B11, ABO, A12, B12 , BO

- 0.3052	0.066	0.5364
- 0.3037	0.276	0.548
- 0.015	0.354	0.649
0.0	0.5	0.3775
0.1345	0.545	0.8245
0.1345	0.545	-0.8245
0.0	0.5	-0.3775

Feed in the king pin inclination and caster angle

9.5   4.0

Position, velocity and acceleration vectors of joints:

A11:

Position vector/-0.9114e+00 0.2530e+00 0.5589e+00/

Velocity vector/-0.4549e-03 -0.2514e-03 0.4732e-04/

Acceleration vector /-0.3132e-03 -0.1773e-03 0.3345e-04/

B11:

Position vector/-0.9667e-01 0.2735e+00 0.6530e+00/

Velocity vector/-0.4835e-03 -0.8308e-04 0.2924e-0e/

Acceleration vector /-0.3421e-03 -0.5347e-04 0.2065e-03/

A12:

Position vector/ 0.1200e+00 0.5201e+00 0.7437e+00/

Velocity vector/-0.3577e-03 0.2795e-04 -0.3165e-03/

Acceleration vector /-0.2519e-03 0.1941e-04 -0.2212e-03/

B12:

Position vector/ 0.1890e+00 0.4947e+00 0.9030e+00/

Velocity vector/-0.4219e-04 -0.5349e-04 -0.3020e-03/

Acceleration vector /-0.2993e-04 -0.3740e-04 -0.2110e-03/

For the right tire

Angular displacement = -0.5619e+00 (-0.3222e+02)

Angular velocity = -0.2649e-02

Angular acceleration = -0.1359e-02

For the left tire

Angular displacement = -0.3602e+00 (-0.2064e+02)

Angular velocity = -0.1628e-02

Angular acceleration = -0.1137e-02

{ Projections of the present configuration of the steering linkage in YZ and ZX planes displayed }

Feed in the camber angle

1.0

Feed in the wheel base

4.565

{ Fig. 2.5 with following values of X_{err}, A_{err} and θ_{err} displayed }

Distance error = Length GH =-0.2507e+01

Area error = Area of triangle EFG = 0.2635e+01

Angle error = Angle EA_OF =-0.4163e+01

Negative distance error implies that the axes of two wheel intersects between front and rear axles.

Negative angle error implies that left wheel has been steered less than its ideal steering value.

Feed in the value of king pin offset and tire width

0.005 0.2

Feed in the load on the steering axle in kgs

500.0

Moment required to turn single stationary wheel

= 0.1166e+03

Internal forces and moments between links :

By link AO-A11 on fixed link:

Force/0.1169e+04 0.1156e+03 0.1497e+03/

Moment/0.3589e+02 -0.4513e+02 -0.2454e+03/

By link A11-B11 on link AO-A11:

Force/ 0.1169e+04 0.1156e+03 0.1497e+03/

By link B11-ABO on link A11-B11:

Force/ 0.1169e+04 0.1156e+03 0.1497e+03/

By fixed link on arm ABO-B11 of link B11-ABO-A12:

Force/ 0.1169e+04 0.1156e+03 0.1497e+03/

Moment/ 0.4562e+01 -0.2649e+03 0.1690e+03/

By arm A12-ABO of link B11-ABO-A12 on fixed link:

Force/ 0.0000e+00 0.0000e+00 0.6315e+03/

Moment/ 0.2842e+02 -0.1165e+03 0.0000e+00/

By link B12-A12 on link B11-ABO-A12:

Force/ 0.0000e+00 0.0000e+00 0.6315e+03/

By link BO-B12 on link B12-A12:

Force/ 0.0000e+00 0.0000e+00 0.6315e+03/

By fixed link on link arm BO-B12:

Force/ 0.0000e+00 0.0000e+00 0.6315e+03/

Moment/0.2842e+02 -0.1165e+03 0.0000e+00/

Moment in link A12-ABO along king pin axis = - 0.1166e+03

Moment in link B11-ABO along king pin axis = 0.2332e+03

Moment required at gear box output = 0.2403e+03

Feed in the steering wheel diameter

0.5

Feed in the value of gear box efficiency

0.8

Steering effort required = 0.2939e+01

In Chapter 5, to study the variations in various parameters, a simplified version of this steering system is analysed for two complete rotations of steering wheel. All results and conclusions are also included in the next Chapter.

Chapter 5

RESULTS AND CONCLUSIONS

5.1 Variations in Tire Rotations, Steering Error and Steering Effort with Steering Wheel Rotation

To study the behaviour of tire rotations, steering error and steering effort as the steering wheel is rotated, the steering system of a truck is analysed, for steering wheel rotation of $-360°$ to $360°$. To illustrate the prominent features in the variations in these variables, the steering system analysed in Section 4.4 has been simplified. Values of the various parameters of the simplified steering system are given below:

Number of universal joints = 0

Type of gear box = 1 (Screw and nut type)

Initial nut displacement = 0.006 m

Length of spindle arm = 0.06 m

Lead of the screw = 0.011 m

gama5 = 0.0

gama6 = 0.0

Type of linkage = 1 (spindle lever linkage)

Coordinates of joints (in meters) :

AO :	- 0.8052	0.066	0.5364
A11 :	- 0.8037	0.276	0.543
B11 :	- 0.015	0.354	0.649
ABO :	0.0	0.5	0.8775
A12 :	0.1345	0.545	0.8245
B12 :	0.1345	0.545	-0.8245
BO :	0.0	0.5	-0.8775

King pin inclination = 0.0

Caster angle = 0.0

Camber angle = 0.0

King pin offset = 0.15 m

Tire width = 0.2 m

Load on steering axle= 500 kgs

Wheel base = 4.565 m

Steering wheel diameter = 0.5 m

Efficiency of gear box = 0.8

Angular velocity of steering wheel = 1.0 rad/s

Angular acceleration of steering wheel = 0.0 rad/s^2

Tables 5.1, 5.2 and 5.3 give the variations in tire rotations, steering error and steering effort with steering wheel rotation.

TABLE 5.1

Tire and Steering Wheel Rotations

Steering wheel rotation (degrees)	Right tire rotation (degrees)	Left tire rotation (degrees)
- 360.0	38.74	53.39
- 315.0	36.74	51.43
- 270.0	34.07	44.97
- 225.0	30.60	38.30
- 180.0	26.20	31.21
- 170.0	25.08	29.57
- 135.0	20.81	23.66
- 90.0	14.50	15.76
- 45.0	7.463	7.776
0.0	0.977×10^{-5}	0.5497×10^{-4}
45.0	-7.564	-7.268
90.0	-14.96	-13.82
135.0	-22.04	-19.57
180.0	-28.75	-24.52
200.0	-31.63	-26.48
225.0	-35.17	-28.74
270.0	-41.47	-32.33
315.0	-47.99	-35.41
360.0	-55.72	-38.09

TABLE 5.2

Steering Wheel Rotation and Steering Error

Steering wheel rotation (degrees)	Distance error (m)	Area error (m^2)	Angle error (degrees)
- 360.0	-1.829	1.055	-9.35
- 315.0	-1.382	0.5181	-5.310
- 270.0	-0.9502	0.2156	-2.708
- 225.0	-0.5028	0.05372	-1.007
- 180.0	-0.4067	0.417×10^{-3}	-0.060
- 170.0	0.0533	0.531×10^{-3}	0.061
- 135.0	0.3884	0.02628	0.2765
- 90.0	0.7541	0.09227	0.2299
- 45.0	0.9967	0.1542	0.07172
0.0	0.0	0.0	0.0
45.0	1.002	0.1556	0.06187
90.0	0.7848	0.09937	0.1791
135.0	0.4720	0.03817	0.2252
180.0	0.1022	0.00193	0.08141
200.0	-0.0729	0.1022×10^{-2}	-0.07008
225.0	-0.2974	0.01789	-0.3530
270.0	-0.7141	0.1143	-1.187
315.0	-1.153	0.3365	-2.615
360.0	-1.661	0.8197	-5.252

TABLE 5.3

Steering Wheel Rotation and Steering Effort

Steering wheel rotation (degrees)	Steering effort (Newtons)	Steering wheel rotation (degrees)	Steering effort (Newtons)
- 360.0	9.391	0.0	19.59
- 315.0	10.84	45.0	18.67
- 270.0	12.49	90.0	17.20
- 225.0	14.29	135.0	15.42
- 180.0	16.13	180.0	13.58
- 170.0	16.53	200.0	12.78
- 135.0	17.82	225.0	11.83
- 90.0	21.56	270.0	11.57
- 45.0	19.75	315.0	8.881
		360.0	7.706

5.2 Discussion on Results

Though desired, for the steering system analysed tire rotations are not symmetric about the point of zero steering wheel rotation i.e. the left tire rotation for any clockwise rotation of steering wheel is different from the value of right tire rotation for the same amount of counter clockwise steering wheel rotation and vice-versa. It is a consequence of the initial nut displacement and the asymmetric nature of the spindle lever linkage. For other types of steering linkages, for instance center point linkage, it is possible to design a linkage, giving symmetric tire rotations. But other design criteria, like convenient gear box position, do not always allow to use such a linkage.

Steering errors in Table 3.2 have the expected trend - in addition to no steering configuration, steering error is zero only for two other configurations ($\sim -175°$ and $\sim 195°$ steering wheel rotations).

Angle error should not be used for comparing errors in two different configurations, because even for comparable steering errors, for smaller steer angles angle error has smaller values. Though values of the angle error relative to the steer angle (i.e. angle error/tire rotation) may be used for the comparison,

distance error and area error should be preferred. But for designing the linkage (to find the scope of improvement) angle error is very important.

In Table 3.3 steering effort is maximum for steering wheel rotation of - 90° and decreases on both sides. This type of torque distribution and the torque distribution shown in Fig. 5.1(a) should be avoided. A good steering design should require almost constant steering effort between extreme wheel position (Fig. 5.1(b)).

5.3 Suggestions for Further Work:

In the present work, major portion of the analysis of a steering system has been presented. It is almost essential to complete this part before a design can be finalized. But to make the design even sophisticated and more reliable, many secondary analyses can also be included. These are suggested below:

1. In addition to the efficiency of gear box, in force analysis, efficiencies of revolute, spherical and universal joints can also be taken into account.

2. Analysis of power steering can also be included. Use of power steering, besides affecting the

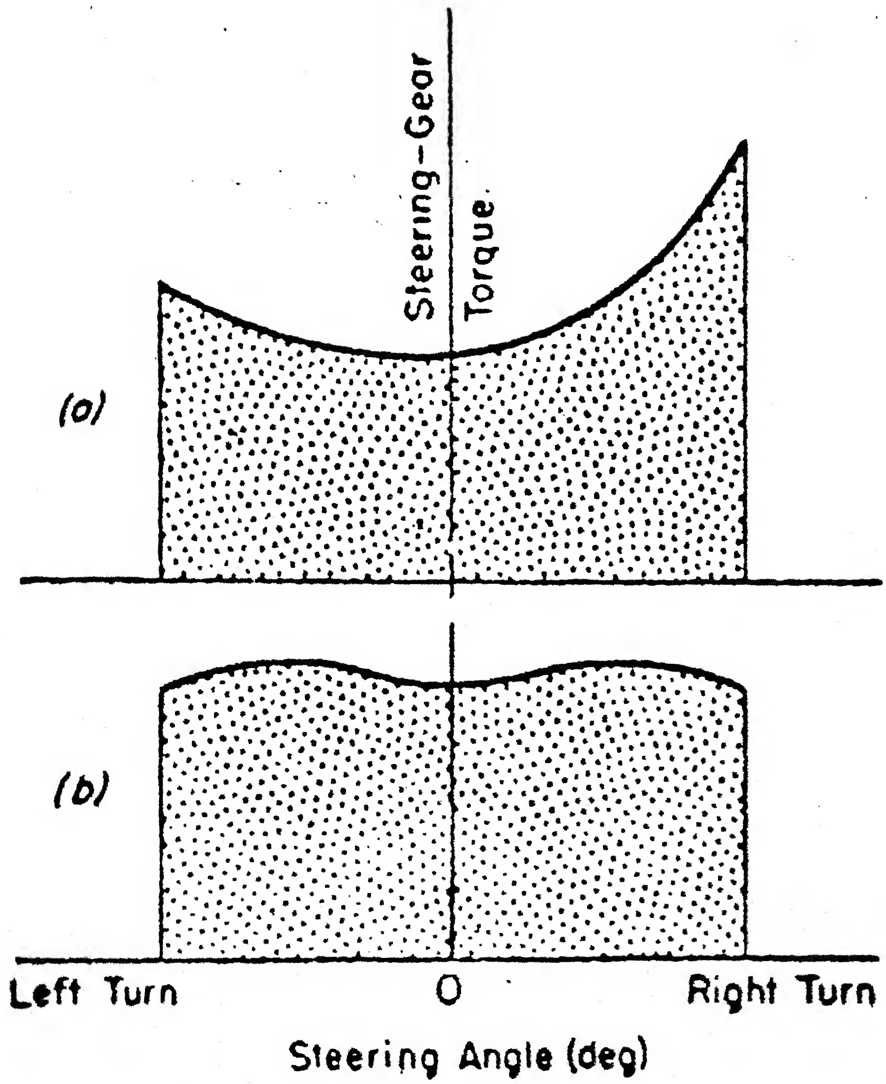


Fig 5.1 Torque on gear box output shaft as a function of steering angle (a) torque distribution for a poor steering design (b) desirable torque distribution.

force analysis, will also require the analysis for variable length of the drag link.

3. Natural frequency of the steering linkage should not match with engine frequency or with road frequency. Cross-sections of various links should be adjusted to avoid the resonance. Therefore if natural frequency of the steering linkages is calculated it will help in finding the cross-sections of links.

4. Because of irregularities in the road, two front wheels move up and down, causing front axle to roll. Axle rolling changes the position and orientation of king pins and steering linkage as a whole. This results in a change in the amount of steering. This error may be calculated. The kick-back felt by the driver, when the vehicle passes over a bump can also be calculated.

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